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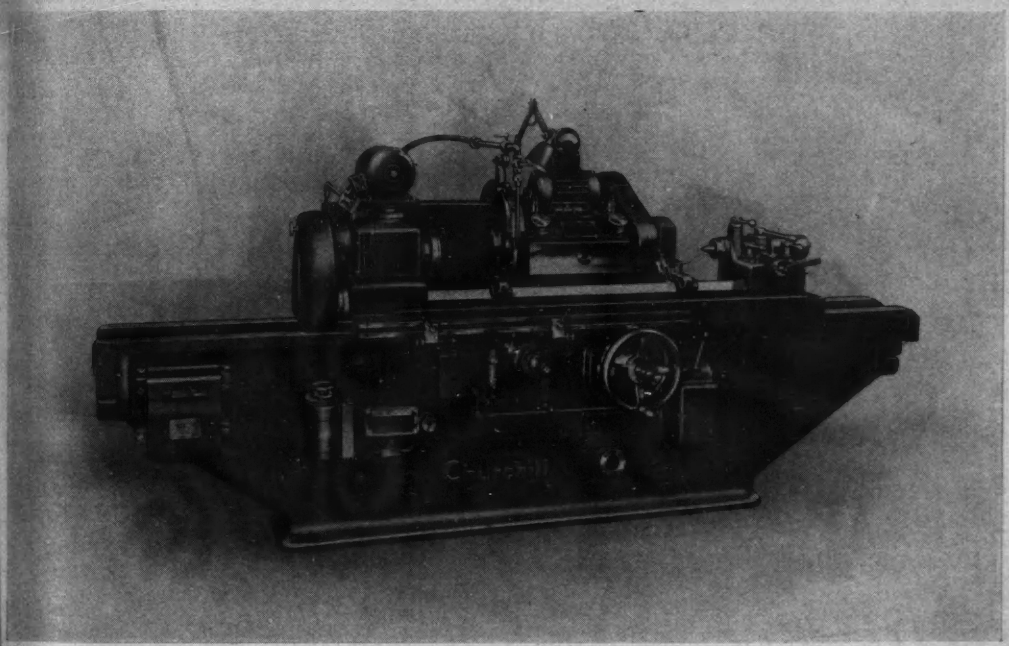

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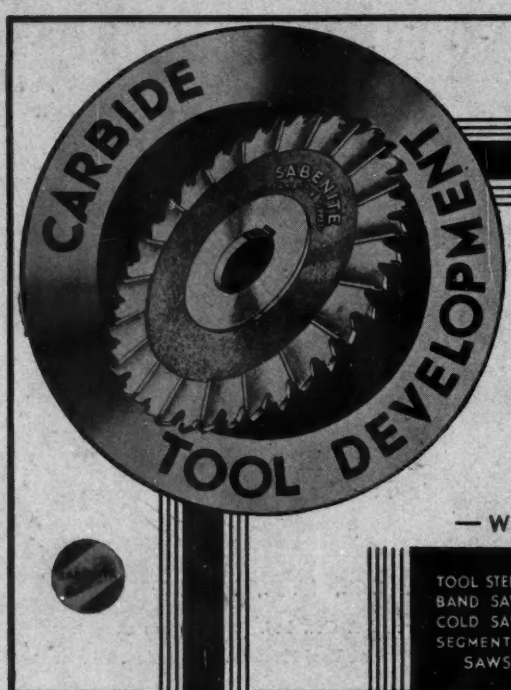

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CONTENTS

page

Aero Engine Governors 301
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 301.

Analytical Determination of the Power Curve of High Speed Road Vehicle Diesel Engines .. 315
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 315.

Boiling of the Liquid Metal in Steel Melting Furnaces 311
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 311.

Centrifugal Casting in the Shipbuilding Industry 321
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 321.

Classified Abstracts 331
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 331.

Concrete and Cement 330
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 330.

Diamond-Pobedite Crowns for Mining Drills 308
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 308.

Electrolytic Cutting of Metal 314
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 314.

Gear Type Pump with Variable Delivery 315
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 315.

Heat Transfer in Air Heaters 318
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 318.

Hydraulic Accumulators for Constant Pressure Operation 328
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 328.

Investigation of Roll Wear in the Hot Rolling of Wire and Strip 322
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 322.

A Liquid Propellant Rocket Motor 329
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 329.

The MeDiCo Mechanical Diagram Corrector 309
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 309.

News of the Month 335
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 335.

Oscillographic Measurement of A.C. Hysteresis Loop 313
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 313.

Scale Glass as a Substitute for Mica 319
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 319.

Thermal De-Gassing of Boiler Feed-Water 325
"*The Engineers' Digest*," Vol. 7, No. 10, *p.* 325.

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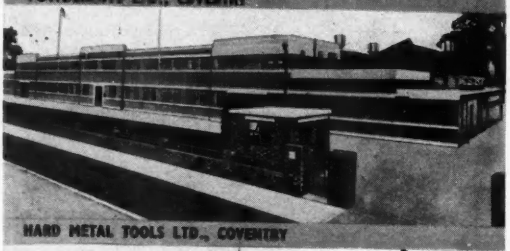
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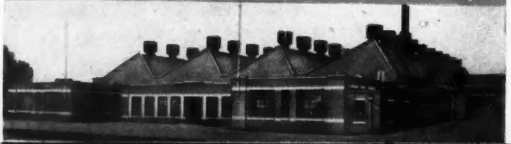
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AIRCRAFT ENGINE GOVERNORS.

By TH. STEIN.* (From *Schweizerische Bauzeitung*, Vol. 127, Nos. 24, 25 and 26, June 15, 22 and 29, 1946, pp. 295-300, 309-313, and 323-327, 36 illustrations.)

1. THE REMOVAL OF THE CONTRADICTIONS BETWEEN THEORY AND PRACTICE.

WHEREAS automatic governing was originally used on stationary power-plant and theoretical work was confined to this sphere, the increasing use of automatic controls usually in connexion with steam plant, presented new problems. When pressure and flow controls were introduced, practice became far in advance of theory. One of the main principles of the earlier theory was overthrown, namely the principle that a control with a servo-motor could not be used without restoring mechanism, owing to lack of stability.

The reason why omission of the restoring mechanism is, in practice, inadmissible with stationary power-plant but in other applications leads to a useful, simplified construction was discovered by the author in 1928 as a mathematical principle known as "self-regulation". Self-regulation occurs even in the governing of stationary power-plant; but to such a small extent that it remained unnoticed and did not incur any noteworthy falsification of the calculated behaviour.

The governing of variable-pitch airscrews was another matter. The fundamentals calculated in 1940 show such a surprising influence of self-regulation that, according to the operating conditions of that time, the control satisfied all requirements without a restoring mechanism. The relationships established were then used to calculate the conditions produced by the Escher Wyss variable-pitch airscrew when used as a landing brake.

Without there being any clue in practice, the theory was produced that, with a continuous increase in the rate of adjustment of the airscrew governor, oscillations must occur because the self-regulating effect is weakened. Here, the governor theory was in advance of practice because in 1943 marked oscillations were experienced with a newly modified airscrew, the oil pumps of which were too large, and these oscillations were damped out, in agreement with the theory, by reducing the rate of pitch variation. Where increased demands require high rates of pitch variation, not only when the blades are in their extreme positions but also for engine-speed control, calculation may indicate the need for a restoring mechanism or an acceleration governor. According to recent publications from the U.S.A., tests on various types of governors have led to the same result. It has been confirmed that the governor theory, supplemented by the law of self-regulation, has removed the apparent contradiction with practice so that a general view can be formed, for future development.

Governors are also used in the jet-propulsion of aircraft. Since here, as in the case of airscrews, the governor theory provides indications as to how the governor is most suitably used, the dynamic relationships between these governor processes are explained in this article.

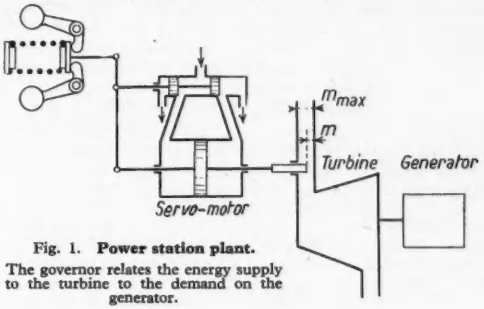


Fig. 1. Power station plant.
The governor relates the energy supply to the turbine to the demand on the generator.

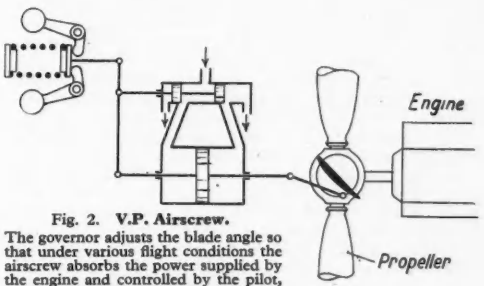


Fig. 2. V.P. Airscrew.
The governor adjusts the blade angle so that under various flight conditions the airscrew absorbs the power supplied by the engine and controlled by the pilot, the engine speed remaining constant. Strong self-regulating effect.

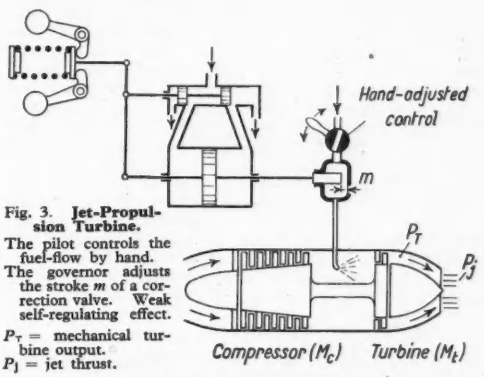


Fig. 3. Jet-Propulsion Turbine.
The pilot controls the fuel-flow by hand. The governor adjusts the stroke m of a correction valve. Weak self-regulating effect.
P_T = mechanical turbine output.
P_J = jet thrust.

Figs. 1 to 3. Diagrammatic comparison of the governors of turbines and aircraft engines

2. DIFFERENCES IN THE EFFECTS OF SELF-REGULATION.

In order to elucidate the similarities, as well as the differences, between the old and new governor problems and to prevent experience with stationary governors leading to incorrect deductions about aero-engine governors, the diagrams of a stationary governor, an airscrew governor and a jet-propulsion turbine governor are reproduced together in Figs. 1 to 3.

In the stationary power-plant, the power output of the turbine is controlled, whereas the current demanded from the generator is not. According to the original theory it was assumed, if the speed were maintained between narrow limits, that slight temporary speed deviations had no effect on the governing process. In fact, this is not so. In order to indicate this effect, let us assume that the power required for the current consumed is proportional to the frequency. If the speed increases, the torque of the generator remains constant. With the setting of the servo-motor unaltered, however, the output of the turbine is practically constant so that torque varies in inverse proportion to the speed. For an opening m of the servo-motor which is too great by, for example, $\Delta m : m_{max} = 1$ per cent compared with the consumption, the speed of the machine would, according to the early theory, increase to infinity, with time, without

*Escher Wyss A.-G., Zürich.

operation of the governor. In fact, however, the acceleration would vanish, owing to self-regulation, as soon as the speed has increased by 1 per cent (the output remaining constant) and the turbine torque had consequently fallen by 1 per cent to reach the correct equilibrium condition. If the self-regulation constant is denoted by :

$$e_s = \frac{\text{percentage decrease in torque}}{\text{percentage increase in r.p.m.}}$$

then under full load $e_s = 1$. As will be seen, self-regulation exerts, in this case, a small stabilizing influence on the governor process at full load. It is, however, a defect affecting every self-regulation process that it diminishes with decreasing load and this has often led to the failure of governors not fitted with restoring mechanisms. If the consumption approaches zero, the stabilizing influence of changes in speed on torque variations disappears.

The self-regulating effect is very much more marked in the case of the airscrew (Fig. 2). Here, the resisting torque increases more rapidly than with the square of the airscrew speed, particularly at high flying speeds, so that, apart from the effects of the light construction, very much smaller acceleration periods of airscrew and engine must be expected. The two together increase the effect of the self-regulation. The difference that the governor does not, in this case, control the energy supply to the engine but the energy expended by the airscrew by adjusting the blade angle is, in principle, unimportant.

In the case of jet-propulsion, the energy supply to the turbine is controlled, as in stationary power-plant, by the governor. If differences in efficiency are disregarded; a variation in speed, with no variation in the servo-motor setting, will, in the case of the stationary turbine cause practically no variation in the turbine output, and in the case of the airscrew no variation in the engine torque. On the other hand, a variation in the speed of the jet-propulsion turbine affects more than the torque of the power-consuming unit, i.e. of the compressor. Even with the fuel supply unaltered, the turbine torque changes because, with varying compressor pressure, a different percentage of the energy of the fuel is applied to the jet energy P_j and the difference is, therefore, not applied to the turbine. For the turbine-compressor combination, it is proposed to take as a basis the characteristics k_μ and k_φ which indicate how the excess moment depends on the governor setting and the speed. It is shown that the magnitude of this torque decreases in simple proportion to increasing r.p.m., as is the case with the stationary turbine. With equal closure times of the governors, the self-regulating effect is greater than it is for the stationary turbine because, owing to the lighter construction, the acceleration period of the jet-propulsion turbine is almost ten times less.

3. PRINCIPLES OF CALCULATION.

The derivation of the governor equations produces the following expressions for the variable-pitch airscrew and the jet-propulsion turbine.

For the assessment of the applicability of governors it has previously been considered sufficient to ascertain whether the governor process is stable, i.e. whether, after a variation in the operating conditions (e.g. from full load to partial load) the deviations from the new equilibrium conditions are damped out, or not. This gives no information, however, about how quickly, and after how many fluctuations, the governor operation will become steady. This is an important point, especially in the case of aero-engines. Theoretically, the duration of the process is infinitely long and it undergoes an infinite number of fluctuations. As useful practical criteria, the damping period $T_{1/10}$ i.e. the time taken for the variation in speed to be damped to 1/10th of its maximum value and therefore to be practically

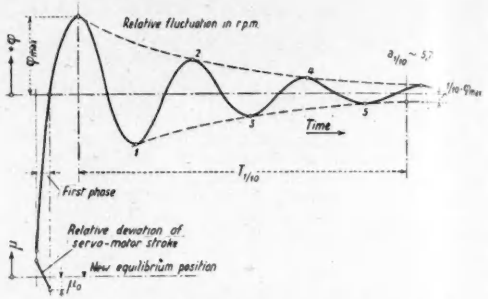


Fig. 4. Simplified assessment of the governor process by the damping period $T_{1/10}$ and the number of fluctuations $a_{1/10}$ until the fluctuations φ are damped to 1/10 of φ_{max}. The displacement of the zero point to the moment when the r.p.m. fluctuation becomes 0 for the first time gives simplified equations for the excess speeds.

entirely damped (Fig. 4), and the number of fluctuations during this period $a_{1/10}$, were introduced. Further, the maximum relative r.p.m. variation φ_{max} (overspeeding) is an important factor for the loading of the engine.

Let $\varphi = \Delta n/n_{0\,max}$ = the relative r.p.m. variation from the new equilibrium r.p.m. n_0 in relation to the maximum value $n_{0\,max}$.

$\mu = \Delta m/m_{max}$ = the relative servo-motor deviation from the new equilibrium position (see Fig. 4) corresponding to the full throttle governor stroke m_{max}.

A. Expressions of General Validity.

For damped oscillations with the characteristic quadratic equation :

$$w^2 + aw + b = 0 \quad .. \quad .. \quad (1)$$

the frequency $$q = \sqrt{b - \frac{a^2}{4}} \quad .. \quad .. \quad (2)$$

the period $$T = \frac{2\pi}{q} \quad .. \quad .. \quad (3)$$

the damping period $$T_{1/10} = \frac{4{\cdot}6}{a} \quad .. \quad .. \quad (4)$$

the number of fluctuations

$$a_{1/10} = 1{\cdot}46\,\frac{q}{a} \quad .. \quad .. \quad (5)$$

the overspeeding for V.P. airscrews with sudden shock loads :

$$\varphi_{max} = f_d \frac{\mu_0}{qT_a} \quad .. \quad .. \quad (6)$$

for jet-propulsion turbines with sudden shock loads :

$$\varphi_{max} = f_d \frac{k_\mu \mu_0}{qT_a} \quad .. \quad .. \quad (7)$$

for jet-propulsion turbines with damped controls :

$$\varphi_{max} = 2f_d \delta \frac{T_s}{T_d} \quad .. \quad .. \quad (8)$$

where f_d = the damping factor (determined from Fig. 5). μ_0 = the ratio of the deviation of the momentary servo-motor setting Δm_0, since the r.p.m. variation φ passes for the first time through 0, to the variation in setting m_{max} of the servo-motor for full throttle torque (see Fig. 4).

$$\mu_0 = \Delta m_0/m_{max}.$$

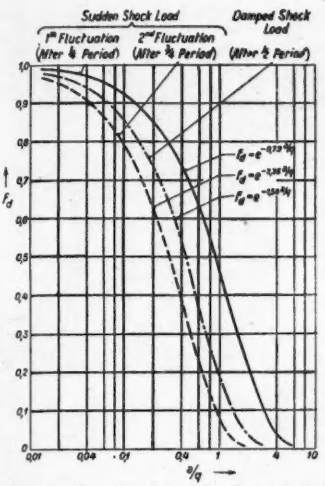


Fig. 5. **Diagram for the determination of the damping factor f_d in the equations of the excess speeds. The equations for the undamped fluctuations are to be multiplied by this factor.**

For a sudden change from idling to full throttle operation, without simultaneous adjustment of the equilibrium speed, $\mu_0 = 1$.

B. Characteristic Governor Equations.

Without restoring mechanism :

$$w^2 + \frac{e_s}{T_a} w + \frac{k}{\delta T_a T_s} = 0 \quad .. \quad (9)$$

with a restoring mechanism :

$$w^2 + \underbrace{\left(\frac{e_s}{T_a} \right.}_{\text{self-regulation}} + \underbrace{\left. \frac{r}{T_s} \right)}_{\text{restoring mechanism}} w + \frac{k}{\delta T_a T_s} = 0 \; .. \quad (10)$$

For V.P. airscrews $k = 1$

For jet-propulsion turbines $k = k_\mu$

C. Governor Constants.

(a) *Self-Regulation Constant e_s and Constant k_μ.*

V.P. Airscrew Generally.

$$e_s = \left(2 + \frac{\Delta\lambda}{\lambda} \right) \frac{z}{z_n} \quad .. \quad (11)$$

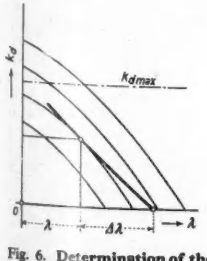


Fig. 6. **Determination of the value $\Delta\lambda$ of the self-regulation constants, which indicates the gradient of the k_d curves.**

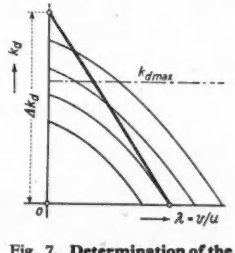


Fig. 7. **Determination of the value Δk_d of the self-regulation constants, for idling speeds.**

is determined from the k_d diagram according to Fig. 6.

V.P. Airscrew, Engine Idling.

$$e_s = \frac{\Delta k_d}{z_n \, k_{d\,max}} \quad .. \quad .. \quad (12)$$

Δk_d is determined from the k_d diagram according to Fig. 7. $k_{d\,max}$ applies to full throttle torque at maximum r.p.m.

Jet-Propulsion Turbine.

For the turbine-compressor combination :

$$e_s = \frac{\text{percentage reduction in torque}}{\text{percentage increase in r.p.m.}} \quad .. \quad (13)$$

for constant setting of the fuel control.

Order of magnitude $e_s \sim 1$

$$k_\mu = \frac{\text{percentage increase in torque}}{\text{percentage increase in fuel consumption}} \quad (14)$$

Order of magnitude $k \sim 0{\cdot}3$

(b) *Restoring Mechanism Constant r.*

For a governor without a restoring mechanism $r = 0$

For a governor with a restoring mechanism $r \sim 1$

(c) *Dynamic Variation.*

$$\delta = \frac{\Delta n_{\max}}{n_{0\,\max}}$$

$\Delta n_{\max}$ = the variation in speed required to open the control valve to produce the maximum rate of adjustment.

$n_{0\,\max}$ = the maximum equilibrium speed.

n_0 = the equilibrium speed which is to be maintained unless it is the same as $n_{0\,\max}$.

(d) *Speed Factor.*

$$z_n = \frac{n_0}{n_{0\,\max}}$$

(e) *Load Factor.*

$$z = \frac{\text{torque at equilibrium}}{\text{torque at full throttle}}$$

D. Time Values.

(a) *Acceleration Period,*

$$T_a = \Theta \frac{\omega_{0\,\max}}{M_{\max}} \text{ in sec.}$$

where Θ = moment of inertia in mkg. s^2

$\omega_{0\max}$ = angular velocity at full throttle in s^{-1}

$M_{\max}$ = full throttle torque in mkg.

Order of magnitude 1 to 2 seconds.

(b) *Closure Period*

T_s = time for adjustment to idling speed from full throttle.

In the case of the V.P. airscrew this is :

$$T_s = \frac{\Delta\alpha_{\max}}{d\alpha/dt}$$

where $\Delta\alpha_{\max}$ = the angle of pitch adjustment required to alter the torque by the full throttle value $M_{\max}$

$d\alpha/dt$ = rate of pitch adjustment in deg./sec.

Order of magnitude of T_s { 0·5 to 6 sec. for V.P. airscrew; 0·1 to 2 sec. for jet-propulsion turbine.

(c) *Damping Period*

T_d = opening time of damping mechanism between control lever and fuel adjustment = time for the adjustment through the full stroke of the fuel valve at maximum rate of adjustment (corresponding to the fully-open position of control valve). Proposed for jet-propulsion turbine.

E. VALUES FOR THE V.P. AIRSCREW.

$$k_d = \frac{2M}{\rho\, u^2\, F\, R}\ ;\ \lambda = v/u$$

M = torque in mkg
ρ = density of air in kg s^2 m^{-4}
u = airscrew tip speed in m/s
R = airscrew radius in m
F = πR^2 airscrew disk area in m^2
v = flying speed in m/s.

4. THE VARIABLE-PITCH AIRSCREW.

These relationships were used in the development of the Escher Wyss variable-pitch airscrew in order to determine the most suitable design.

(a) *Self-Regulation and Restoring Mechanism.*

A governing process is good if its damping period $T_{1/10}$ is short and the number of fluctuations $a_{1/10}$ is small. This is so, in both cases, if the factor a in the denominator of equations (4) and (5) is as large as possible. According to equations (1) and (10), however,

$$a = \underbrace{\frac{e_s}{T_a}}_{\text{Self-Regulation}} + \underbrace{\frac{r}{T_s}}_{\text{Restoring Mechanism}}$$

The self-regulating effect predominates over the

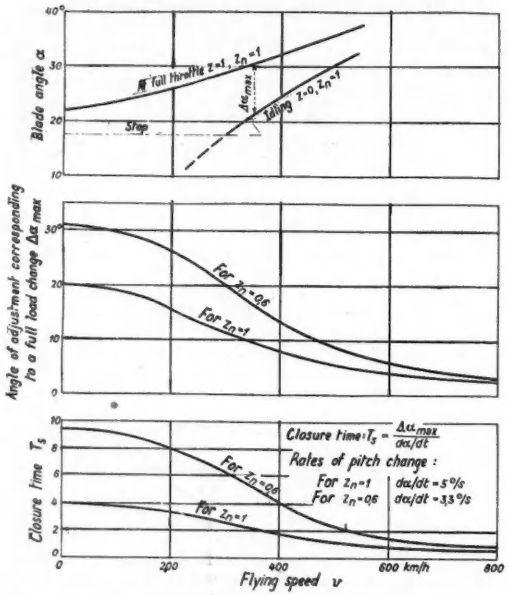


Fig. 8. **Determination of the closure time T_s of the servo-motor for the adjustment of the blade angle by the full load value $\Delta\alpha_{max}$ for various r.p.m. values. In contrast with stationary governors, the closure time is not constant but varies.**

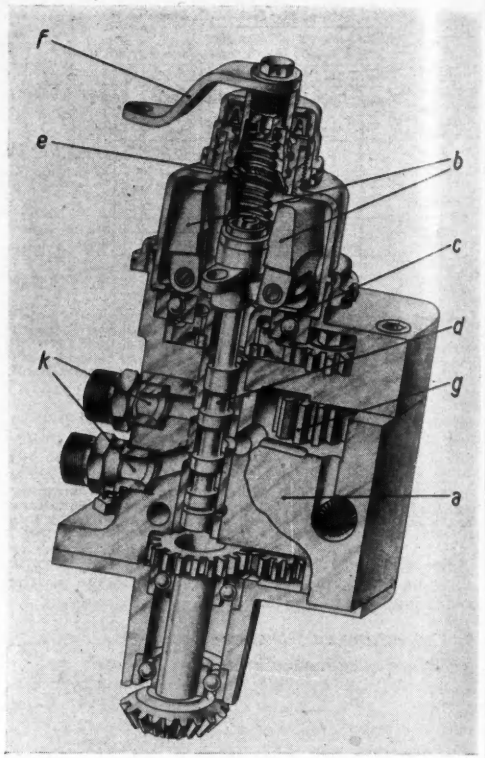


Fig. 9. **The Escher Wyss V.P. airscrew governor.**
(*a*) casing, (*b*) centrifugal pendulum, (*c*) governor bearing, (*d*) distributor valve, (*e*) return spring for r.p.m. adjustment, (*f*) spring tension adjusting lever, (*g*) gear-type oil pump, (*k*) connexions to servo-motor.

effect of the restoring mechanism if :

$$e_s\, T_s/T_a > r$$

where $r \sim 1$ for a rigid restoring mechanism coupling. For an acceleration period T_a = 1·6 sec. in low-speed flight the closure time T_s is, according to Fig. 8, much greater i.e. $T_s/T_a > 1$. In high-speed flight, e_s assumes considerably greater values, as in Fig. 14. If the rate of adjustment is not increased, that is, if the closure time T_s is not reduced, $e_s\, T_s/T_a$ may be much greater than 1 so that the self-regulating effect may be much greater than the restoring mechanism effect. If, however, a restoring mechanism must be used owing to increasing demands, then it is effective only when the closure time is much weakened. It is only possible, therefore, to increase the effectiveness of self-regulation by slow pitch adjustment, and that of the restoring mechanism by rapid pitch adjustment.

(b) *Normal, Powered Flight.*

Since a governor is most easily constructed without a restoring mechanism as no connexion is then required between the servo-motor and the governor, an attempt was made in the development of the Escher Wyss governor to avoid great rates of adjustment of pitch. On the other hand, the maximum rate of pitch adjustment (up to 50 deg/sec.) is required for air-screw braking. It was important, therefore, to determine, from the expressions obtained, when it is possible to avoid the use of restoring mechanisms and by what means the governor process can be improved. Even at idling speeds, when for other applications, governors without restoring

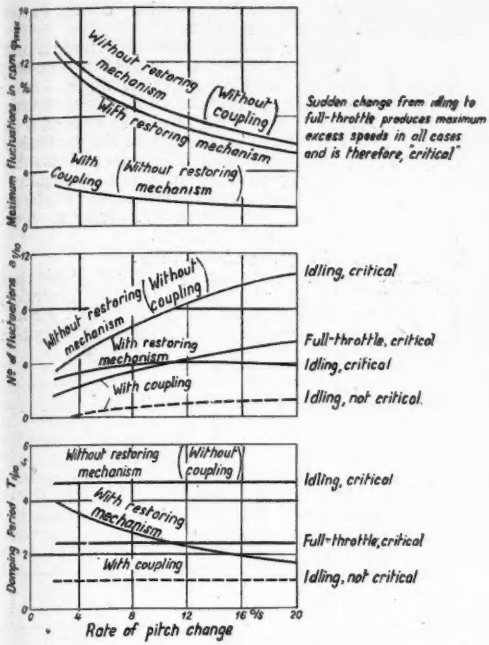


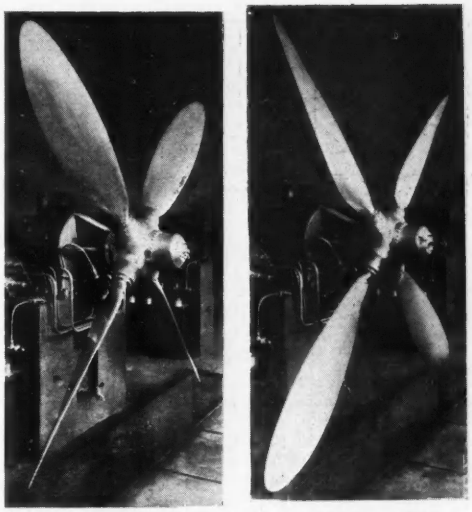
Fig. 10. Reduction of excess speed of V.P. airscrew by increased rate of pitch change using a restoring mechanism or a coupling between the throttle lever and the governor. The use of the coupling produces fewer and smaller fluctuations than does the restoring mechanism (calculated for 300 km/h = minimum flying speed at low engine speed, with governor operating).

mechanisms fail, the self-regulating effect is sufficient because at low flying speeds (below 300 km/h) the airscrew operates as a fixed-pitch airscrew without governing. If it were merely a question of reducing the damping period $T_{1/10}$ and the number of fluctuations $a_{1/10}$ no restoring mechanism would be necessary and it would be sufficient simply to use moderate rates of pitch adjustment (Fig. 10). On the other hand, a restoring mechanism may be used to reduce the excess speed φ_{max}.

The reduction of the excess speed, produced by a restoring mechanism, for the same rate of pitch change is unimportant (Fig. 10). The circumstance is important, however, that the restoring mechanism allows of increasing the rate of pitch adjustment which reduces the excess speed, whereas the governor without a restoring mechanism would allow the number of fluctuations $a_{1/10}$ to increase to inadmissible values (Fig. 10, centre). A governor with a restoring mechanism reduces the number of fluctuations with increasing rate of pitch change and they are not allowed to increase. As shown by the curves for $T_{1/10}$ in Fig. 10, the governor process then reaches equilibrium in a much shorter time.

(c) *The Coupling of the Governor and the Throttle Lever.*

A very considerable reduction in the excess speed can, however, be obtained without a restoring mechanism and with moderate rates of pitch adjustment if the equilibrium speed is set at a lower value by the throttle lever, at partial loads. This is effected in the case of the Escher Wyss V.P. airscrew, by a combined control lever, at the same time reducing the strain on the pilot. In order to be able to calculate the maximum variation in r.p.m. φ_{max} it is necessary to ascertain the value assumed by the servo-motor variation μ_0 when the r.p.m.



Figs. 11 & 12. Normal position and braking position of Escher Wyss airscrew. In the braking position the sharp trailing edge, and not the rounded leading edge, is directed forwards (clockwise rotation).

variation φ has decreased for the first time to zero in the first governor phase. The value $\mu_0 = 0{\cdot}22$, found by calculation, signifies that the r.p.m. variation is, in this case, reduced to 22 per cent of the value obtained without the coupling. Fig. 10 shows that, with the coupling, even for moderate rates of pitch adjustment, much lower excess speeds are produced than with a restoring mechanism and without the coupling, for the same rates of pitch adjustment.

With the coupling for the reduction of the excess speeds, however, it is possible to increase the rate of pitch adjustment by 10 deg/sec before more fluctuations are produced than with uncoupled governors with restoring mechanisms. Owing to the increased self-regulating effect at moderate engine speeds, the governor functions practically periodically ($a_{1/10} \sim 0$ to 1, Fig. 10) when the engine is idling. In this case, the self-regulating factor applying to full load is still predominant and at maximum r.p.m. is greater than for idling when, without the coupling, the number of fluctuations is critical.

For normal powered flight, therefore, a coupling is better for the reduction of the excess speed than a restoring mechanism.

(d) *Governing When Using the Airscrew as a Landing Brake.*

Compared with the fixed blade position, the landing brake effect may be increased because the pitch is controlled to give constant engine speed. The governor must control the pitch in the opposite way from that in normal flight so that, with increasing engine speed and therefore excess engine torque, the blade angle becomes steeper ; the resisting moment of the airscrew increases and the speed again falls to the required value.

If, in practice, it is only required to govern for full braking and constant engine speed, it is sufficient to investigate the full-throttle values. It is recognized from the shape of the k_q curves in the negative pitch range (Fig. 13 dotted lines) that they increase with increasing λ instead of falling as for normal flight (continuous lines). Thus, for full throttle operation according to equation (11) ($z = 1$, $z_n = 1$), the self-regulation constant for the normal flight is $e_s > 2$ and for the airscrew braking range $e_s < 2$.

The self-regulating effect for airscrew braking is much less pronounced than that for normal flight

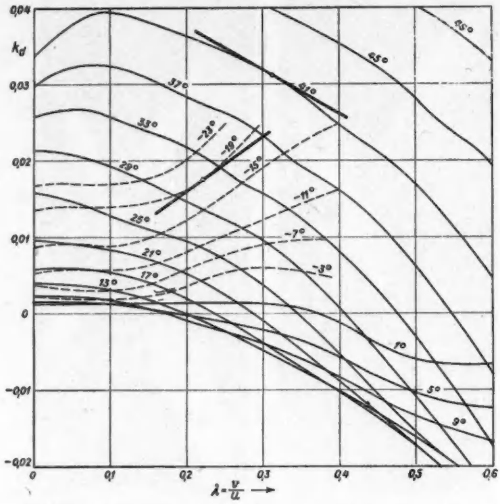


Fig. 13. The tangents to the k_d curves, which are influenced by the self-regulating constants, decrease for positive pitch angles (normal flight) with increasing λ so that the self-regulating effect is increased. Conversely, the tangents increase, for negative pitch angles (braking) so that the self-regulating effect is weaker than in normal flight.

because the headwind works against it instead of favouring it.

(c) *Braking in Flight.*

The governor behaves quite differently when using the airscrew as a brake during flight. The governor must, in this case, also operate at partial loads. Towards the idling speeds, as Fig. 14 shows, the self-regulating effect is less than at full-throttle. Further, there is the danger that, in full-throttle braking flight, the engine might stall. In order to avoid the possibility of the engine being turned backwards by wind-milling, it is necessary to have very high rates of pitch adjustment, particularly in high speed flight. If the governor is to prevent this reverse rotation automatically if the engine fails, it must operate with a greater rate of pitch change, which weakens the self-regulating effect. This effect, however, is already weak, as shown by Fig. 14. It is further reduced by great rates of pitch change, whereas the effect of a restoring mechanism increases. In many cases which can be determined mathematically, it is therefore more suitable to use a restoring mechanism or the acceleration governor preferred in the U.S.A.

5. JET-PROPULSION TURBINE.

(a) *The Purpose of the Governor.*

A free choice of engine speed by variation of the blade angles to suit any particular power is not, in contrast to the V.P. airscrew, possible in the jet-propulsion turbine. Here the governor usually affects the flow of fuel, as shown in Fig. 3, and the engine speed is self-adjusting. Nevertheless, the governor, apart from its function of preventing excess speeds, provides a useful indication of the output. In addition to simple governors for limiting the maximum engine speed (Fig. 15), governors of complicated construction, with constant-speed restoring mechanisms, are built.

It is not, however, sufficient when a governor is used merely to limit the speed and to control only one unit, simply to interrupt the fuel supply, as would a safety valve, when a certain speed is exceeded. As soon as the limiting speed is reached the governor must control the jet-propulsion turbine in such way that the fluctuations in speed remain within permissible limits and that they are damped quickly without dangerous "hunting". The relationships derived will allow to make preliminary calculations for each individual case, in order to choose the most suitable governor constants.

(b) *The Function of the Closure Time.*

In the jet-propulsion turbine, only one needle valve or adjustable member of the fuel pump is controlled, so that closure times of fractions of a second are possible. In stationary power plant such high rates of adjustment

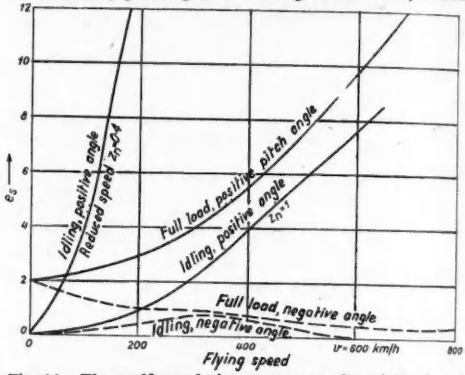


Fig. 14. The self-regulation constant for full throttle operation is greater than 2 for positive pitch angles, and less than 2 for negative angles. At idling speeds, with negative pitch angles, there is almost no self-regulating effect because the head-wind counteracts this effect in braking flight. A restoring mechanism or an acceleration governor is, therefore, required in braking flight.

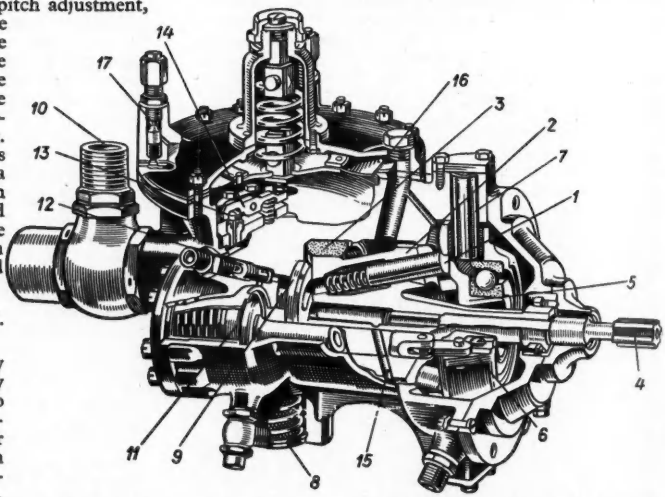


Fig. 15. The speed-limiting governor made by Lucas for the Rolls-Royce Derwent jet-propulsion turbines.
1. pump plunger. 2. rotor. 3. carbon bush. 4. driving shaft. 5. cam plate. 6. control ring. 7. its axis of rotation (trunnion). 8. oil inlet filter. 9. ported disk controlling inlet and outlet of the pump plungers. 10. outlet connexion. 11. pressure control piston. 12. restriction orifice. 13. plate valve. 14. lever controlling the governor. 15. radial passages. 16. governor diaphragm. 17. air bleed valve.

are used only for auxiliary controls and not for the main servo-motor. Moreover, the governor and the servo-motor for the fuel supply are close together so that a mechanical restoring mechanism can easily be fitted.

It is proposed now to examine how fluctuations may be avoided by suitable selection of the closure time for a governor with a restoring mechanism. A fully aperiodic governor operation is achieved if, as in (10)

$$e_s \underbrace{\sqrt{\frac{T_s}{T_a}}}_{\text{Self-Regulation}} + r \underbrace{\sqrt{\frac{T_a}{T_s}}}_{\text{Restoring Mechanism}} = 2\sqrt{\frac{k\mu}{\delta}}$$

The effect of the closure time on self-regulation is inversely proportional to that on the restoring mechanism. Since the acceleration period T_a of the jet-propulsion turbine is almost ten times less than that of a stationary turbine, the self-regulating effect is stronger for the same closure time and affects the governor process to such an extent that, for closure times of one second or more, this effect cannot be ignored (Fig. 16). Without the self-regulating effect the number of fluctuations would increase very quickly to inadmissible limits, with increasing closure time, as shown by the dotted line. In fact, however, the increasing self-regulating effect is so strong that with longer closure times the number of fluctuations is reduced and the aperiodic range is almost reached (only one fluctuation which is damped to 1/10). From the aspect of " hunting " any closure time would, therefore, be permissible.

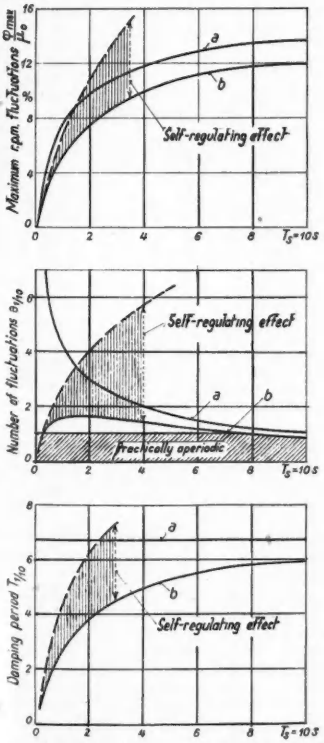


Fig. 16. **Small closure times T_s accentuate the effect of the restoring mechanism and conversely, with increasing T_s, the self-regulating effect is so increased that governors without restoring mechanisms (simplest form of constant-speed governors) may be used when not subject to excessive initial fluctuations.**

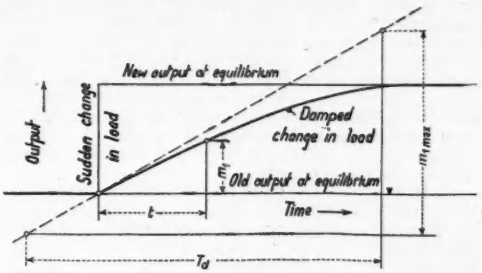


Fig 17. **Determination of the damping period T_d for the calculation of the governor curve and the much reduced excess speed when the throttle lever is damped and the rate of increase is limited independently of the pilot.**

On the other hand, Fig. 16 shows that the excess speeds increase, for closure times of 1 to 3 seconds, to values which are greater than those of stationary turbines. This is accentuated by the fact that the acceleration period T_a is almost ten times less, owing to the light construction. According to the arrangement used, however, the initial value μ_0 of the servo-motor deviation from the equilibrium position can be very small. If the error in the manual adjustment is 5 per cent and the free motion of the governor corresponds to 10 per cent of the full-throttle torque, then $\mu_0 = 5 + 10 = 15$ per cent. Then, according to Fig. 16, a closure time of 2 seconds, for example, is permissible. The greatest demands are satisfied by governors with restoring mechanisms, the closure times of which amount to a fraction of a second.

(c) *The Attainment of Complete Combustion.*

The small combustion chambers of the jet-propulsion turbine, which are subject to more than ten times the combustion chamber loading per cubic metre capacity than the oil-burning furnaces of stationary steam generators, are able to withstand this only by reason of the much greater quantity of excess air passing through the turbines than through the furnaces. An increase in the loading, due to an increase in the quantity of fuel, must

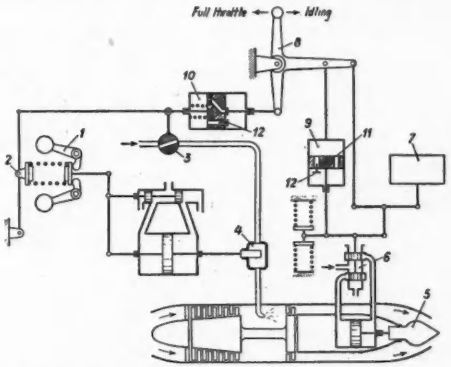


Fig. 18. **Proposed governor process for a jet-propulsion turbine with temporary increase in the excess air for improving the engine response.**

1. Centrifugal pendulum. 2. r.p.m. adjustment. 3. valve for coarse adjustment. 4. fine adjustment of the fuel-flow. 5. variable jet-nozzle. 6. control valve for jet-nozzle. 7. barometric control. 8. throttle lever. 9. device to increase the air-flow by temporarily increasing the jet aperture when first the throttle is fully opened (inoperative when moving to idling position). 10. damping deviation to delay the increase in the fuel-flow when the throttle lever is moved to the fully open position (inoperative when moving to idling position). 11. throttling orifice. 12. non-return valve.

not be too rapid since the air-flow increases only with an increase in the compressor speed. The quantity of excess air must not be allowed to become too small because complete combustion would not then be possible in the highly loaded chamber. On the other hand, the reduction to smaller powers may be relatively rapid as the full air-flow is still provided by the still unchanged speed, when the fuel quantity is reduced, so that an increase in the excess air is produced which is permissible for the combustion.

If it is required to limit the rate of increase of the load, independently of the pilot, a damping device may be fitted between the pilot's control lever and the fuel adjustment. Equation (8) makes it possible to calculate whether a governor without a restoring mechanism, as the simplest form of constant-speed governor, will allow excess speeds or not.

Instead of preventing an excessive lag in the increase in speed (air-flow), compared with the fuel-flow, merely by damping the increase in loading, it is possible directly to control the speed. For the case of a jet-propulsion turbine operating with a jet-pipe of variable cross-section (Fig. 18), it will be shown how the engine response can be increased by an additional temporary control of the speed. The control of the jet-pipe cross-section is normally used to improve the operating conditions of the compressor by adjusting them to the power, altitude, and speed of flight. Parallel with this control, the jet-pipe cross-section may be increased by a control unit, the motion of which is conveyed temporarily by the throttle lever, so that a larger part of the energy of the fuel is used for accelerating the turbine, at the cost of the energy in the jet, so that the air-flow for the combustion of the increased fuel supply is increased. The fuel-flow is controlled by a damping device (10) which, at the end of the damping period, in contrast to the damping unit (9) for the jet-pipe area, provides a continuous and not a temporary adjustment. To insure that the increase in the air-flow is sufficient when the fuel-flow increases, the damping unit (10) for the fuel may be made to operate more slowly. The arrangement should be such that, when the power is reduced, the damping unit is ineffective. The diagram serves only to indicate the basic elements for the improvement of the governor process if maximum demands are to be made of it.

RUSSIA

DIAMOND-POBEDITE CROWNS FOR MINING DRILLS.

By H. N. TEGKAYEV. (From *Gornii Journal*, No. 3, March, 1946, pp. 25-26. 2 illustrations.)

THE Sadonsky mines had been using, both for exploratory work and for production, the Swedish "Krelius" diamond drills which were not standing up to the work. The diamond crown wore rapidly, and with the change in its diameter the diamonds became unduly exposed and eventually lost, causing costly replacements. The drills worked in fits and starts, requiring constant expert attention and the cores came out badly or not at all.

The diamonds were exposed, not only by wear on the periphery, but also by the washing away of the end face between them by the action of the flushing stream which issued through the gaps under high pressure (5 to 9 atmos., 71 to 128 lb./sq. in.), carrying with it highly abrasive fragments of ore. The original crown, indicating typical wear, is shown in Fig. 1.

To overcome this difficulty a new type of crown was designed, as shown in Fig. 2, in which the diamonds are interspersed with blocks of "pobedite" (a Russian extreme-hardness alloy) and the water is released through radial holes just before reaching the end face. Provided the holes are not too far back, sufficient water gets to the cutting face to cool it and keep it from clogging.

With a crown ring of 35 to 45 mm. (1⅜ in. to 1¾ in.) six or eight industrial diamonds, of 0·75 carats each, are fixed in the usual way in a lead seating, flush with the end face. A similar number of hexagonal or octagonal blocks of pobedite are located between the diamonds and protrude by ·25 to ·5 mm. (·010 ins. to ·020 ins.) beyond the end face. Some are nearer the outer, and some are nearer the inner periphery, so as to protect both from undue wear.

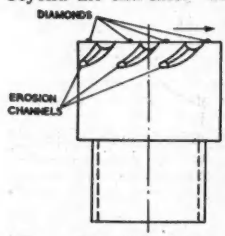


Fig. 1. **Diamond crown**

After a year's experience the following results have been recorded :—

1. The expenditure of diamonds, when boring through highly fissured rock, has fallen to between 12 and 19 per cent.

2. The crown gives 28 times longer service.

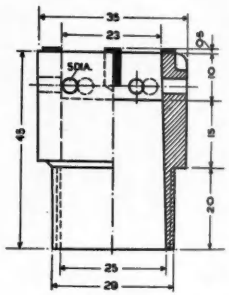
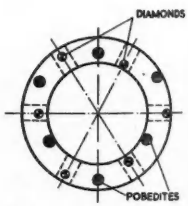


Fig. 2. **Diamond—Pobedite crown**



3. Even poor quality diamonds can now be utilized.

4. The drill runs smoothly, because the diamonds do not dig in so much.

5. The core comes out more freely (aided by the fact that the inner bore tapers out from 23 to 25 mm. (·9 in. to 1 in.).

6. Less expert labour can be used because there are fewer troubles and breakdowns.

7. There is no need to start the bore with a jumper drill, as the new crown operates very smoothly.

8. The new crown is 15 to 20 per cent slower in action when actually boring, but so much unnecessary lifting and lowering is cut out that the overall monthly performance has risen by 30 to 40 per cent.

9. The amount of re-boring required on unsatisfactory bores is reduced from 35 to 15 per cent.

THE "MeDiCo" MECHANICAL DIAGRAM CORRECTOR.

By L. BIENFAIT. (From *Revue Générale des Chemins de Fer*, Vol. 65, No. 3, May-June, 1946, pp. 75-80, 9 illustrations.)

IT is claimed that the use of the apparatus described in this article, eliminates much of the work normally involved in determining the best alignment of curves on permanent way rail tracks by calculation, and at the same time gives better results in a much quicker time.

These claims are supported in a foreword by M. Leduc, the Engineer in Chief to the *Service Central des Installations fixes et de la Construction* of the S.N.C.F., for which service the apparatus is under consideration.

For many years, track engineers have tried different methods for obtaining from measured offsets, the slews required to get the best alignment of curved tracks. The speeding up of train services has emphasised the importance of constantly maintaining proper alignment.

The various means in current use are basically much alike, but none are free from methods of trial and error. Furthermore, fairly lengthy calculations are involved and the successful application depends on considerable individual experience.

The diagram correcting apparatus "MeDiCo" described in the article is essentially a calculating machine which produces, in visible form, a diagram of the corrected slews, both positive and negative, as measured from offsets on the track.

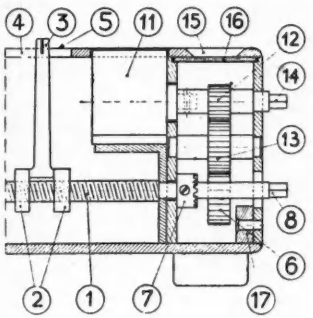


Fig. 1.

Each of the offsets is measured on site and is reproduced on the MeDiCo (see Figs. 1 and 4) by the medium of an endless screw or worm (1) which, by its rotation, causes a lateral movement of a screw nut (2) and moves a sliding index or indicator (3) which runs in a sight-slot (4). A millimetre scale (5) is provided adjacent to the slot and the worm (1) has at one end a gearwheel of 30 teeth (6) and on the other a driving coupling (7).

The gears are staggered on opposite sides of each successive worm, all of which have squared ends to fit a crank-handle (4) which is provided for rotation. This rotation moves the index (3) so that a curve diagram appears on the surface of the machine.

A second crank-handle (Fig. 2) has a pinion (9) with 15 teeth to mesh with gear wheel (5) and the end of the crank-handle (10) engages the coupling (7) of each worm.

This crank-handle can be placed on the end of the worm to engage with the coupling and laterally with the gear wheels (6) of adjoining offsets.

By turning the crank-handle, the index of the worm with which it is engaged is moved in the direction and to the extent required, but the indices of the two adjoining worms move by only half the amount in the opposite direction.

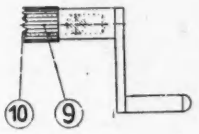


Fig. 2.

The original plotted offsets are then adjusted as desired.

Succeeding improvements to the diagram are made till the fresh offsets forming the required diagram are obtained.

The machine also gives the corresponding slews at each point without additional operation. This important result is obtained in the following manner :— (See Fig. 3—half cross section).

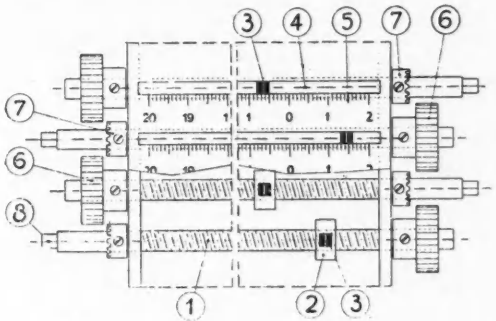


Fig. 3.

The index indicating each offset by the manipulation of three worms undergoes two motions : first, those of value $\pm d$ and secondly those of value $\pm \frac{d}{2}$ resulting from the movement of one of the two adjoining offsets. The final slew for each point is the algebraic sum of the motions of the corresponding index at that point. The slew is recorded and indicated as follows :—

Above each worm near the pinion there is a revolution counter (11) which records the movement marked on the corresponding index (3).

The ratio between the number of teeth in the pinion thus engaged (Fig. 1 (12) and (13) and Fig. 2 (9)) and the thread of the worm is such that the rev-counter shows unity when the index moves 1 mm. The rev-counter works in both directions, adding in one direction and subtracting in the other. Each counter can be operated singly for resetting to zero without alteration to the index of the corresponding offset.

Through the sight-hole (Fig. 1 (15)) the number of the peg corresponding to the offset can be seen, the number being inscribed on a sliding scale (16).

The prototype machine provides for 32 offsets of 250 mm. (200 mm. for the positive and 50 for the negative offsets). The apparatus is 39 cm. in length and 38 cm. in width and weighs 6 kg. A box with handle is provided for the machine.

Since an alignment requiring amendment can always be sectioned by an element of 30 pegs of 250 mm. it has

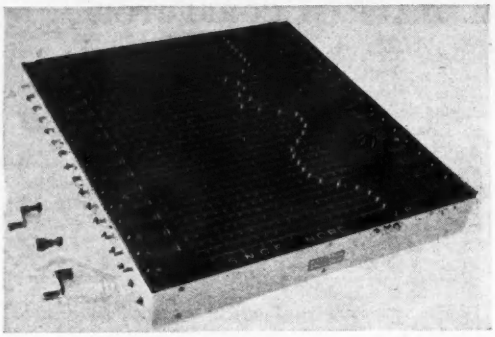


Fig. 4.

been found possible to deal satisfactorily with all cases in practice.

In making the calculations on the machine for correcting alignment, the following operations are performed successively :

All counters are set to zero by rotating each counter rod by the crank-handle. To set up the diagram the scales are adjusted to reproduce the values of the pegs on the track. The index of each offset is moved to the extent and in the direction required by turning the worm with the crank-handle.

The amendment of the group of three successive offsets can then be made.

It is sufficient to know how to read a diagram for the operator to gradually improve its shape as it forms on the machine.

The amended offsets are shown by the index rod above the corresponding millimetre scale. Slews are given by the counters of each offset, negative slews being indicated by the figures appearing and positive slews by the subtraction of the number shown from 10,000.

The machine can deal with all alterations to alignments capable of solution by calculation including transitions, negotiation of the points, limited radii and general and detailed improvements.

Fig. 5 shows the amended diagram of a corrected alignment produced on the machine. The indices show the amended offsets, the counters give the corresponding slews and the original diagram is also shown.

Comparative results obtained by calculation methods and on the MeDiCo machine respectively show very similar diagrams, but in the typical example shown in the original text the average slew by the former method is 75 mm. as against only 19 mm. by the machine.

The author remarks that an assistant often works all day on a job involving an adjustment of 50 pegs and requires four or five trials to reach a satisfactory solution. He claims that in an analogous case, a better result can be obtained with the MeDiCo calculating machine in less than an hour. It is capable of popularizing and simplifying to the limit the alignment of curves by the method of correction of offsets. Its use on the service of rail road for the whole of the S.N.C.F. is under consideration.

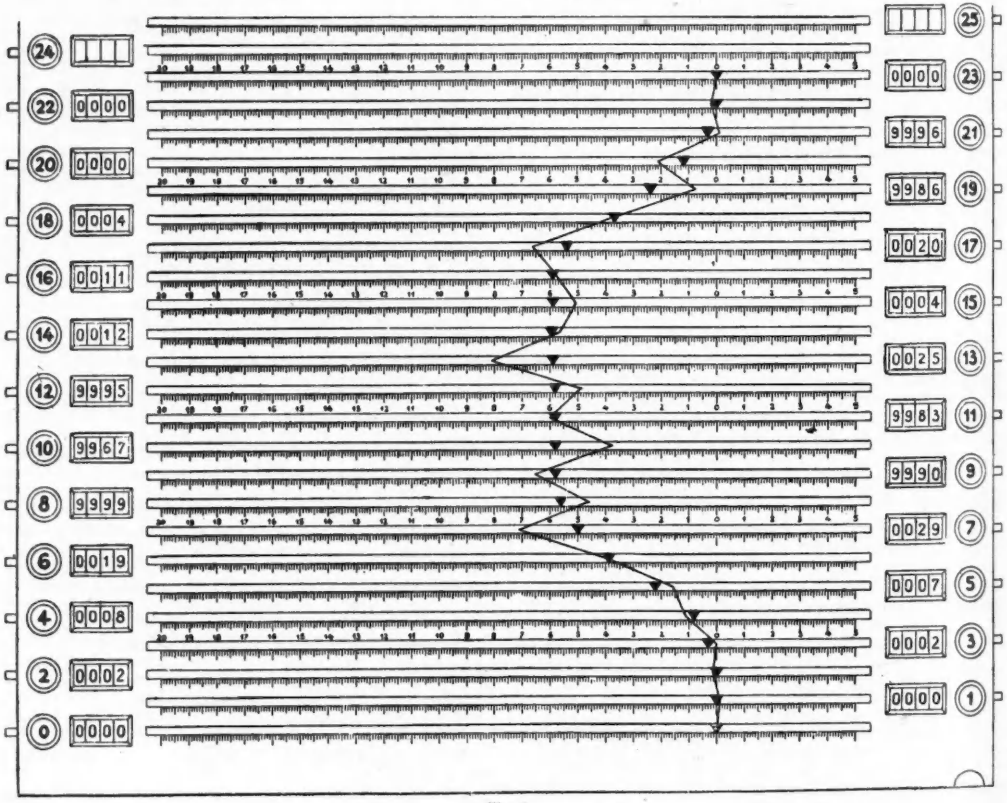


Fig. 5.

BOILING OF THE LIQUID METAL IN STEEL-MELTING FURNACES.

By V. S. KOTSHO. (From *Stal*, No. 2/3, 1945, pp. 55-60, 2 illustrations.)

THE boiling of the liquid metal exerts an important influence upon the technological and thermal processes taking place in the steelmaking furnace. Boiling assists the circulation of the metal in the bath and promotes the heating of the metal ; it accelerates the reactions and also equalizes the composition and the temperature of the bath. The energy required for the circulation of the bath is supplied from the gases which rise in the bath in the shape of bubbles. A computation of the mechanical work involved in the boiling process, as it takes place in an open hearth furnace, was first carried out by H. Schenk*. However, as shown by the investigations of the author, Schenk's computations are incorrect, as the latter erroneously assumed that the entire energy of the gases is absorbed by the slag and the metal. Actually the greater part of this energy is not absorbed by the metal but by the atmosphere. Schenk also made the erroneous assumption that the energy absorbed by the metal does not depend upon the depth of the bath. He assumes that the mechanical energy available in the boiling process, is determined by the product of the volume of the gases developed and rising in the bath per unit of time and the average pressure of the gases in the bath, so that it can be written :

$W = V_{\text{pav, t}}\, P_{\text{av}}$, where W is the mechanical energy, $V_{\text{pav, t}}$ is the volume of gases rising in the bath per unit of time, and P_{av} is given by

$$P_{\text{av}} = P_{\text{metal}} + P_{\text{slag}} + P_{\text{atmosphere}}$$

Actually, the metal and the slag do not absorb this energy entirely, but only that portion given by the product of gas volume and metal and slag pressures, so that $W_1 = V_{\text{pav, t}}\,(P_{\text{metal}} + P_{\text{slag}})$, while the portion absorbed by the atmosphere is given by

$$W_2 = V_{\text{pav, t}}\, P_{\text{atmosphere}}$$

It can therefore be concluded that the mechanical energy absorbed by metal and slag is considerably smaller than that contained in the gases. Furthermore Schenk completely neglects to take into account the work of gas expansion performed during the ascent of the bubbles, and this is so large that it must be allowed for.

THE WORK PERFORMED BY A SINGLE BUBBLE RISING IN THE BATH.

Before discussing the entire energy involved in the boiling process, the amount of energy absorbed by the liquid metal from a single gas bubble rising in the bath and emerging from it will be investigated ; and in order to simplify mathematical treatment it will be assumed that the bath is composed only of liquid metal, no slag being present. The findings arrived at can then be applied in analogous manner to the rising of a bubble in a layer of slag. There exist two different ways of approach to the analytical treatment of the problem. In the first, it is assumed that the volume of the bubble expressed in terms of N.P.T. remains unchanged during the ascent of the bubble in a bath, composed solely of liquid metal (Fig. 1). As it can also be assumed that the gas content of the bubble and the liquid metal will be in the state of thermal equilibrium, the expansion of the bubble, as it rises, can be treated as a process of isothermal expansion.

The work performed by the metal in lifting the bubble can be found by multiplying the lifting force acting upon the bubble by the vertical distance through which the bubble has travelled. The lifting force will depend upon the difference between the density of the liquid metal and that of the gas contents of the bubble,

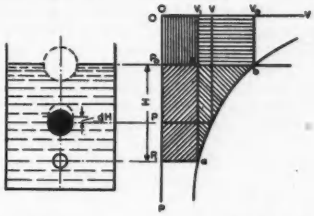


Fig. 1. **Diagram of gas condition in bubble (1st case).**
$a - b$ = Hyperbola for $PV = Po\,Vo$ = constant

and it will also depend upon the volume of the bubble. Both the volume of the bubble and the density of its gas content are functions of the pressure exerted by the liquid metal upon the bubble, that is, functions of the static head and of the temperature of the liquid metal.

The work performed is given by

$$dA = F\,dH, \text{ where}$$

$$F = V_0 \frac{P_0}{(P_0 + H\gamma_{\text{metal}})} \cdot \frac{T}{273} \left[\gamma_{\text{metal}} - {}^{\text{o}}\gamma_{\text{gas}} \frac{(P_0 + H\gamma_{\text{metal}})}{P_0} \cdot \frac{273}{T} \right]$$

$$\text{and } dA = V_0 \frac{P_0}{(P_0 + H\gamma_{\text{metal}})} \cdot \frac{T}{273} \left[\gamma_{\text{metal}} - {}^{\text{o}}\gamma_{\text{gas}} \frac{(P_0 + H\gamma_{\text{metal}})}{P_0} \cdot \frac{273}{T} \right] dH \quad (1)$$

and where dA = work performed
F = lifting force acting upon the gas bubble
dH = distance travelled by the gas bubble
V_0 = volume (N.P.T.) of the gas contained in the bubble
P_0 = atmospheric pressure
H = total distance travelled by the gas bubble
γ_{metal} = specific gravity of the liquid metal
T = absolute temperature of the liquid metal
${}^{\text{o}}\gamma_{\text{gas}}$ = specific gravity of the gas (N.P.T.) contained in the bubble.

The total work performed over the distance H is obtained by integrating equation (1) which yields :

$$A = \int V_0 \frac{P_0}{(P_0 + H\gamma_{\text{metal}})} \cdot \frac{T}{273} \left[\gamma_{\text{metal}} - {}^{\text{o}}\gamma_{\text{gas}} \frac{(P_0 + H\gamma_{\text{metal}})}{P_0} \cdot \frac{273}{T} \right] dH + C$$

It has been found by experiment that the temperature T of the liquid metal remains practically constant throughout the bath during the boiling process, and V_0, P_0, γ_{metal} and ${}^{\text{o}}\gamma_{\text{gas}}$ may therefore also be taken as constants. It can thus be written :

$$A = V_0 P_0 \frac{T}{273} \gamma_{\text{metal}} \int \frac{dH}{P_0 + H\gamma_{\text{metal}}} - V_0 {}^{\text{o}}\gamma_{\text{gas}} \int dH + C$$

The ntegration constant is easily found for $H = 0$ and $A = 0$; it therefore is

*H. SCHENK : *Physics and Chemistry of Metallurgical Processes*, Moscow, 1936.

$$A = V_0\, P_0 \frac{T}{273} \ln\left(1+\frac{H\gamma_{metal}}{P_0}\right) - V_0\, \overset{0}{\gamma}_{gas} \cdot H \qquad (2)$$

In order to explain the physical meaning of this expression, the problem will now be analysed in a different manner. The right hand side of Fig. 1 shows the condition of the gas contained in the bubble, charted in a V-P diagram. The shaded areas of the diagram represent the work performed by the gas in the bubble. The amount of mechanical work transmitted from the bubble to the liquid metal is obtained as the difference between A_0' and A_0''. Here A_0' is the work performed by the expansion of the gas from the pressure P_1 to P_0 as represented by the points P_1—a—b—V_0 in Fig. 1, while the work A_0'' refers to the stage at which the bubble has pierced the surface of the metal. This stage is represented by the points P_0—c—b—V_0 in the diagram, the pressure having the constant value P_0 correlated with the atmospheric pressure.

The difference between these two amounts of work is represented by the diagram area P_1—a—b—c—P_0—P_1. Since

$$A_0' = P_1\, V_1 \int_{V_1}^{V_0} P \cdot dV$$

and

$$A_0'' = P_0 \cdot V_0$$

it follows that

$$A_0 = A_0' - A_0'' = P_1\, V_1 + \int_{V_1}^{V_0} P \cdot dV - P_0\, V_0$$

But as it is also $P \cdot V = P_1\, V_1 = P_0\, V_0 = \text{const.}$

$$\text{it follows } A_0 = \int_{V_1}^{V_0} P \cdot dV$$

$$\text{which yields } A_0 = P_0\, V_0 \ln\left(1+\frac{H \cdot \gamma_{metal}}{P_0}\right) \qquad (3)$$

The expansion of the gas proceeds at constant temperature T, so that equation (3) which is based upon N.P.T. conditions can also be written

$$A = P_0\, V_0 \frac{T}{273} \ln\left(1+\frac{H\gamma_{metal}}{P_0}\right) \qquad (3a)$$

By comparing formula (3a) with formula (2), it is found that the former differs from the latter only with respect to the term $V_0 \cdot \overset{0}{\gamma}_{gas} \cdot H$ which represents the work performed by lifting the bubble by the vertical distance H. Actually, however, this term is so small in comparison with the first term of the expression that it can be neglected altogether. It therefore follows that the work performed in bringing the gas bubble to the surface is equal to the work of gas expansion transmitted to the liquid metal.

Furthermore, equation (3) determines the diagram area enclosed by the isotherm and by the ordinates $V = V_1$ and $V = V_0$. This diagram area therefore represents the entire work of isothermal expansion from pressure P_1 to pressure P_0. It is well known that in the case of isothermal expansion the entire amount of heat supplied is converted into work, and it can therefore be concluded that the mechanical work expended on the circulation of the liquid metal exactly equals the heat transferred from the liquid metal to the gases leaving the bath under conditions of isothermal expansion. Or, in other words, the entire amount of heat absorbed by the gas bubbles is converted into work employed for the circulation of the liquid metal.

The other way of approach to the analytical solution of the problem is based upon the following concept. A very small bubble of gas immediately upon generation begins to ascend in the liquid metal, with the volume of gas contained in the bubble continuously increasing during the upward movement of the bubble. This increase in volume is due to the reaction C + FeO → CO + Fe which takes place upon the surface film of the bubble. Furthermore the bubble which is filled with carbon monoxide will absorb such nitrogen and hydrogen as is contained in the liquid metal.

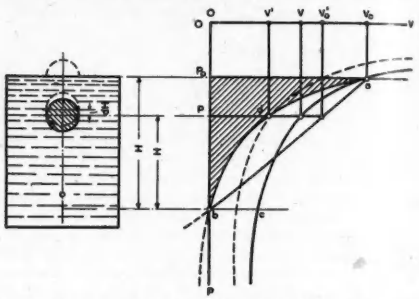


Fig. 2. **Diagram of gas condition in bubble (2nd case).**
a — c = Hyperbola for $PV = Po\ Vo$ = constant
a — b = Variation in bubble volume (on N.P.T. basis) with height H.
a — d — b = Hyperbola characterizing actual change in gas condition in bubble.

The variation in the gas volume of the bubble (reduced to N.P.T. conditions) with the height H can follow any kind of functionality. In the simplest case (Fig. 2) the functionality may be assumed to be of the form

$$V_0' = V_0 \frac{H}{H_1} \qquad (4)$$

It will then be $dA = F\, dH$ where

$$F = V_0 \frac{H}{H_1} \frac{P_0}{(P_0+H\gamma_{metal})} \frac{T}{273} \left[\gamma_{metal} - \overset{0}{\gamma}_{gas} \frac{(P_0+H\gamma_{metal})}{P_0} \cdot \frac{273}{T}\right]$$

so that

$$dA = V_0 \frac{H}{H_1} \frac{P_0}{(P_0+H\gamma_{metal})} \frac{T}{273} \left[\gamma_{metal} - \overset{0}{\gamma}_{gas} \frac{P_0+H\gamma_{metal}}{P_0} \cdot \frac{273}{T}\right] dH$$

and therefore

$$A = \frac{V_0\, P_0}{H_1} \frac{T}{273} \int \frac{H}{(P_0+H\gamma_{metal})} \left[\gamma_{metal} - \overset{0}{\gamma}_{gas} \frac{(P_0+H\gamma_{metal})}{P_0} \frac{273}{T}\right] dH + C$$

and by integrating :

$$A = V_0\, P_0 \frac{T}{273} \left[1 - \ln\left(1+\frac{H\gamma_{metal}}{P_0}\right) \frac{P_0}{H \cdot \gamma_{metal}}\right] - V_0 \overset{0}{\gamma}_{gas} \frac{H}{2} \qquad (5)$$

This expression determines the amount of work exerted by the rising bubble upon the liquid metal. It is now necessary to establish the amount of work supplied from the metal to the gas content of the bubble and absorbed in the process of the expansion of the gas contents of the latter. In Fig. 2, this amount of work is represented by the shaded area of the diagram. In order to be able to compute this area, the mathematical function expressed by the expansion curve must be known. For this purpose it will be assumed that the

expansion of the gas be isothermal and that the volume increase (in terms of N.P.T.) be given by expression (4). Under these conditions the curve *a*—*b* will express the relationship

$$P\,V = \frac{H}{H_1} P_0\,V_0.$$ It will therefore be

$$dA_0 = \frac{P_0\,V_0}{H_1} \cdot \frac{H}{P} \cdot dP$$

and

$$A_0 = P_0\,V_0 \left[1 - \ln \left(1 + \frac{H\gamma_{metal}}{P_0}\right)^{\frac{P_0}{H \cdot \gamma_{metal}}}\right] \quad (6)$$

or

$$A = P_0\,V_0 \frac{T}{273}\left[1 - \ln \left(1 + \frac{H\gamma_{metal}}{P_0}\right)^{\frac{P_0}{H \cdot \gamma_{metal}}}\right] \quad (6a)$$

A comparison of formulæ (5) and (6) shows that in analogy with the comparison carried out in the case of the first variant, the two formulæ differ only by the term $V_0\gamma_{gas}\frac{H}{2}$. This term represents the amount of work performed by lifting the bubble by the vertical distance H. As was already stated, this work can be neglected.

So far only the work performed by a single bubble has been considered, and the investigation must now be extended to an analysis of the work of circulation performed by all the gas bubbles present. According to equation (3) the total work performed by all the gas bubbles generated per unit of time will be

$$A_{total} = V_0 P_0 \frac{T}{273} \ln \left(1 + \frac{H\gamma_{metal}}{P_0}\right) \quad .. \quad (7)$$

and according to equation (4) it will be

$$A_{total} = V_0 P_0 \frac{T}{273}\left[1 - \ln \left(1 + \frac{H\gamma_{metal}}{P_0}\right)^{\frac{P_0}{H \cdot \gamma_{metal}}}\right] \quad (8)$$

where V_0 is now considered to represent the total volume of all the gas bubbles. With regard to the depth H, the effective depth H_{eff} will be assumed to equal 30 to 40 per cent of the total depth of liquid metal.

Formulæ (7) and (8) can be written in the form

$$W_1 = K \cdot \frac{\Delta C}{\Delta \tau} \cdot Q \cdot T \cdot \ln \left(1 + 0{\cdot}1 \frac{H_{eff} \cdot \gamma_{metal}}{P_0}\right) \quad (9$$

$$W_2 = K \cdot \frac{\Delta C}{\Delta \tau} \cdot Q \cdot T \left[1 - 10 \frac{P_0}{H_{eff} \cdot \gamma_{metal}} \cdot \ln \left(1 + \frac{H_{eff} \cdot \gamma_{metal}}{P_0}\right)\right] \quad .. \quad .. \quad .. \quad (10)$$

where W is the mechanical work performed by the process of boiling, given in kW, $K = 0{\cdot}116$ a conversion factor, $\Delta C/\Delta\tau$ the rate at which the chemical combination of the carbon proceeds, expressed in per cent C per minute, Q the weight of the charge in tons, T the absolute temperature of the liquid metal in deg. K., H_{eff} is the effective depth of the metal bath in metres, γ_{metal} the density of the liquid metal in kg per litre volume, P_0 the atmospheric pressure in kg per sq. cm.

With the following numerical values of

$\frac{Q}{T} = 70$ tons / 1893 deg. K. $\quad \Delta C/\Delta\tau = 0{\cdot}005$ per cent C/min $\quad H_{eff} = 0{\cdot}14$m

$\gamma_{metal} = 7$ kg per litre

we obtain

$W_1 = 0{\cdot}116.0{\cdot}005.70.1893.0{\cdot}095 = 7{\cdot}33$ kW

$W_2 = 0{\cdot}116.0{\cdot}005.70.1893.0{\cdot}05 = 3{\cdot}84$ kW

This compares with a value of 76·2 kW as found with the use of Schenk's erroneously conceived formula.

SWITZERLAND

OSCILLOGRAPHIC MEASUREMENT OF A.C. HYSTERESIS LOOPS.

By A. GOLDSTEIN. (From *The Brown Boveri Review*, Vol. 32, No. 7, July 1945, pp. 242-246, 7 illustrations.)

THE technical a.c. magnetizing curve $B_{max} = f(H_{eff})$ for ferro-magnetic materials, derived from measurement of the current and voltage, may prove quite adequate in the majority of cases. In others, however, e.g. for magnetization with various frequencies and different current and voltage wave-forms, it is imperative that the characteristic of the hysteresis loop be known. The article describes a method of determining a.c. hysteresis loops under the above conditions which requires no special measuring equipment. The method can also be adopted to determine $B_{max} = f(H_{max})$ in the case of ferro-magnetic materials with high flux densities, which otherwise is difficult due to the distortion of the current and voltage curves.

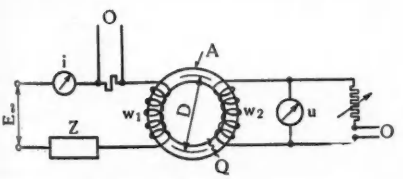


Fig. 1. **Circuit for oscillographic measurement of alternating current hysteresis loop.**

A Test core.
w_1 Current winding of test core.
w_2 Voltage winding of test core.
Z Series-connected impedance.
O Oscillograph leads.
$E_{\sim}$ A.C. supply voltage.
i Current in test coil.
u Voltage across test coil.

The test-circuit is shown in Fig. 1. The impedance Z is provided to adapt the characteristic of the voltage u across the test coil to the conditions actually occurring in practice. Fig. 2 shows an example of an oscillogram for medium flux density. To enable the hysteresis loop to be determined, the flux density (B) in the core under test must be known for every instantaneous value of the magnetizing ampere-turns per cm. length of path of the magnetic circuit (H) throughout the a.c. cycle. The oscillograms, however, only give relative instantaneous values of the current i and voltage u for the windings w_1 and w_2 of the test coil.

The transition from u to B is given by :

$$u = \frac{d\Psi_2}{dt} = w_2 Q \frac{dB}{dl} \cdot 10^{-8} \text{ volts} \quad .. \quad (1)$$

where Ψ_2 is the flux linkage of winding w_2 and Q the cross section of the test core. From equation (1), the flux density is :

$$B = B_0 + \frac{10^8}{w_2 Q} \int_{t_0}^{t} u\,dt \quad .. \quad .. \quad (2)$$

The value of H is obtained from the current in the

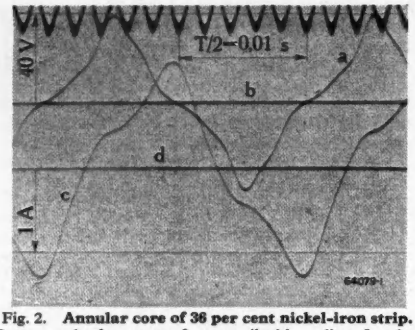


Fig. 2. Annular core of 36 per cent nickel-iron strip.
Current and voltage traces for test coil with medium flux density at 50 cycles.
(*a*) Voltage u across winding w_2 of test core.
(*c*) Zero line of voltage.
(*c*) Current i in winding w_1 of test coil.
(*d*) Zero line of current.

usual way by

$$H = \frac{i \cdot w_1}{l} \text{ AT/cm} \quad .. \qquad (3)$$

Fig. 3 is a practical analysis of the oscillograms in Fig. 2. The voltage curve is divided into $n(12)$ straight line parts and the resulting areas F_k ($k = 1, 2, ... n$) measured ; the areas represent flux changes $\Delta\Psi$, with the corresponding changes in flux density

$\Delta B_k = \frac{10^8}{\omega_2 Q} . F_k$ and the flux density B_m at point m is

$$B_m = \frac{10^8}{w_2 Q}\left[-\frac{1}{2}\sum_1^{n/2} F_k + \sum_1^{m} F_k \right] \text{ gauss} \quad .. \qquad (4)$$

the corresponding magnetizing force is :

$$H_m = \frac{i_m w_1}{l} \text{ AT/cm.} \qquad .. \qquad (5)$$

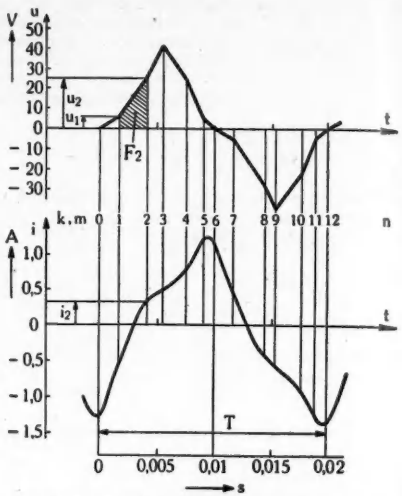


Fig. 3. Analysis of oscillogram (Fig. 2).
The voltage-time surfaces F_k are calculated and added together. The sum gives the flux density B_m at time point m. The corresponding magnetizing force H_m is proportional to the current i_m.

where i_m is the current at point m.

A somewhat different procedure is recommended for tests with high flux densities. Only the point corresponding to i_{max} is taken from each oscillogram. The value B is obtained by integration of the voltage u during the half-wave preceding the current peak, whence

$$B = \frac{1}{2}\frac{10^8}{w_2 Q}\int_0^{T/2} u\,dt \text{ gauss} ; H \doteq i_{max}\frac{w_1}{l} \text{ AT/cm} \quad (6)$$

To obtain the curve $B = f(H)$ several oscillograms must be recorded with various voltages and currents in the test coil.

RUSSIA

ELECTROLYTIC CUTTING OF METAL.

K. Y. SKORNIAKOV. (From *Promishlennaya Energetika*, No. 1, January, 1946, p. 11.)

A NEW way of cutting metal by a combination of electrical and mechanical action has been introduced at the Stalin works in the town of Gorki. It takes the place of circular saw work and is particularly suitable for heat-treated steels, hard alloys and thin tubes, but is also useful for ordinary sections.

From a table of typical results given below it will be seen that one electrolytic saw is able to do the work of several ordinary ones.

Section	No. of cuts per hour.		Increase in production
	By circular saw	By electrolytic saw	
¼in. × 1in.	60	240	4-fold
1·2in. dia.	40	120	3-fold
0·8in. dia.	30	180	6-fold
¼in. × ¼in.	100	1500	15-fold

The construction is as follows. The cutting element is a smooth disk, ·025 in. to ·040 in. thick, the diameter of which, according to the sizes of metal to be cut may be up to 12 in. It is mounted similarly to an ordinary saw but rotates at 2,000-3,000 r.p.m., and is connected to the positive side of a dynamo. The metal to be cut is connected to the negative side and is insulated from the disk mountings.

A special electrolyte is pumped through a system of pipes to the point where cutting is to take place. The rotating disk is then pressed against the work by the action of a counterweight and the electric current, in its passage from the work through the electrolyte to the disk, rapidly removes the metal by electrodynamic and electrochemical action.

The voltage required is 18 to 30 volts and the current, according to size of section, varies from 50 to 100 amps. Thus a d.c. welding set can be used to provide the necessary current.

This method can be employed on sections up to 2 in. thick. The speed of cutting varies greatly with thickness: with 1¾ in. thickness the feed is ½ in./min., with ⅝ in. thickness—12 in./min., while ¼ in. bronze can be cut at 28 in./min. This illustrates to what extent the speed of cutting depends upon the current density.

Despite the heavy currents used, there is a net saving of power on the cutting operations which, in the case of the works in question, is estimated to be 140,000 KWh. per year effected by the use of two electrolytic saws.

RUSSIA

ANALYTICAL DETERMINATION OF THE POWER CURVE OF HIGH-SPEED ROAD VEHICLE DIESEL ENGINES.

By G. W. MELNIKOFF. (From 3rd Technical-Scientific Conference of the Kalinin-Polytechnic Institute, Leningrad, September, 1944, pp. 39-41.)

THE suitability of a Diesel engine for a given transport job is best judged by the power curve of the engine. Formulæ have been given by Poincet and Khlistoff, for calculating power curves of petrol engines, but they cannot be used for Diesel engines. The author suggests the following equation for calculating the b.h.p. of a high-speed Diesel engine :

$$N_e = \frac{427}{60 \cdot 75 . m} g_t \cdot \eta_H \cdot H_u \cdot \eta_t \cdot \eta_0 \cdot \eta_m \cdot n \cdot i \text{ H.P.}$$

where m is the "cycle factor",

g_t . . . theoretical amount of fuel supplied by the fuel pump in kg./cycle,
η_H . . . volumetric efficiency of the fuel pump
H_u . . . lower calorific value of fuel in Cal./kg.,
η_t . . . thermal efficiency of cycle,
η_0 . . . relative efficiency,
η_m . . . mechanical efficiency,
n . . . crankshaft speed, r.p.m., and
i . . . number of cylinders.

The volumetric efficiency of the fuel pump η_H, for a given fuel system, is entirely determined by the speed of the engine crankshaft. For the other efficiencies, η_t, η_0 and η_m, empirical formulæ are offered. For the thermal efficiency, based on data tabulated by F. Schmidt, the following expression is suggested :

$$\eta_t = 0 \cdot 1675 \sqrt[4]{\alpha \frac{P_{max}}{P_1}},$$

where α indicates excess air, and $\frac{P_{max}}{P_1}$ the ratio of maximum combustion pressure over initial pressure ($P_1 \cong 1$ atm.). After a few modifications and after introducing empirical relations of n for η_v, $\frac{P_{max}}{P_1}$, η_0, and η_m, the initially given equation is transformed (a) for $n_{nom.} \leqq 1600 - 1700$ r.p.m. ; into :

$$N_e = \frac{N_{e\,nom}}{n_{nom}} n \left(1 \cdot 3 - 0 \cdot 17 \frac{n^3}{n^3{}_{nom}} - - 0 \cdot 13 \frac{n_{nom}}{n} \right) \sqrt{\frac{\eta_h}{\eta_{h\,nom}}} \text{ H.P.}$$

(*b*) for $n_{nom} \geqq 1700$ r.p.m. :

$$N_e = \frac{N_{e\,nom}}{n_{nom}} n \left(1 \cdot 25 - 0 \cdot 2 \frac{n^3}{n^3{}_{nom}} - - 0 \cdot 05 \frac{n_{nom}}{n} \right) \sqrt{\frac{\eta_H}{\eta_{H\,nom}}} \text{ H.P.}$$

In these equations $\eta_{H\,nom}$ is the volumetric efficiency of the fuel pump at the nominal engine speed. The formulæ can also be used for petrol engines, if the performance of the carburetter, i.e., the fuel volume supplied by the fuel jets at varying engine speeds, is known. Data calculated with the new formulæ have been compared with performance data of 16 high-speed Diesel engines and 4 petrol engines and showed fair agreement.

A power curve obtained by means of the new formula, e.g. in the case of a projected engine, may be used as the basis for calculating the road performance of the whole vehicle. On the other hand, it may be used for drawing the curves of characteristic factors of engine performance, such as b.m.e.p., fuel consumption and efficiency, in relation to engine speed. The formulæ also permit, without bench test, of checking the suitability and efficiency of the fuel pump of a given engine.

The equations given above show that, of all factors affecting the power curve of a Diesel engine, the most important are the volumetric efficiency of the fuel pump (η_h) and the correct choice of injection timing.

ITALY

GEAR TYPE PUMP WITH VARIABLE DELIVERY.

By R. CHIAPPULINI. (From *L'Ingegnere*, Vol. 20, No. 4, April 1946, pp. 254-260, 16 illustrations.)

POSITIVE displacement type pumps for adjustable delivery, such as are used for industrial purposes, are exclusively of the reciprocating piston or sliding vane type. Pumps of these types which can also be used as hydraulic motors are, however, somewhat complicated both in construction and operation ; and it is for this reason that the author proposes a simpler pump design than any hitherto known.

As will be seen from Fig. 1, the delivery of fluid by a gear type pump is due to the gear teeth of one of the gears entering into the liquid filled spaces between the gear teeth of the other gear wheel. In pumps of this kind there exists an inherent possibility of a reflux or shortcircuiting of the liquid to the suction side through the gear clearances at the faces as well as at the tips of the teeth.

In the case of the pump design shown in Fig. 2, the follower gear is made axially slidable for the purpose of delivery control. Here, however, the gear faces A and B are obviously ineffective from the aspect of delivery, and the apparently effective length C of the gear is laterally in communication with the voids V_1 and V_2. This possibility of reflux is particularly clearly seen in the sections I and II given in the lower part of Fig. 2.

Referring to Fig. 3, it will be appreciated that the reflux will be considerably reduced if the voids V_1 and V_2 are eliminated by providing side flanges which follow the lateral displacement of the gear when it is laterally shifted for the purpose of adjusting the delivery rate of the pump. It will, however, also be seen that while the elimination of the void V_2 is easy enough, the elimination of void V_1 presents a much more difficult stereometric problem. To be exact, the left hand side flange piece, besides having the shape of a piston with a diameter equal to the extreme diameter of the gear wheel, must also be shaped to envelope a portion of the periphery of gear 1. In addition to this, a sealing ring concentric with the axis of the driving gear No. 1 must also be provided, this sealing ring being forced onto a seat in the

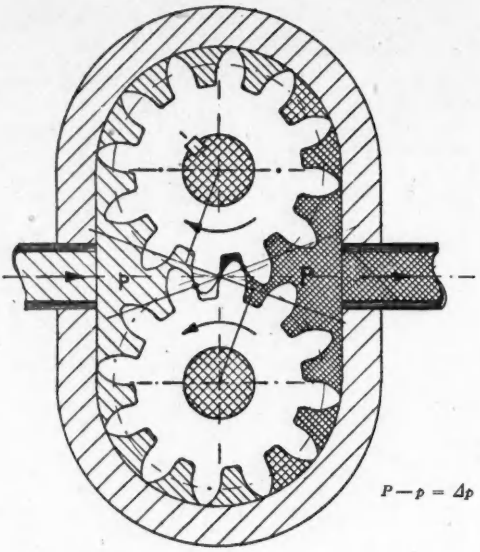


Fig. 1.

wall of the pump housing. Or, as an alternative, the gear may be equipped with a solid rim to take the place of the sealing ring.

In this way a reflux can take place only along the labyrinth path, the flow area of which is given by the voids between the mutually engaged teeth of the two gears. Obviously, the flow in the direction of the path of contact cannot be important, considering that at least one pair of teeth is always in contact, as shown in Fig. 4. It is for this reason that the pump shown in Fig. 3 is free from reflux in the direction of the path of contact of the teeth. In order to keep the residual reflux as small as possible, the play between the two gears should be as

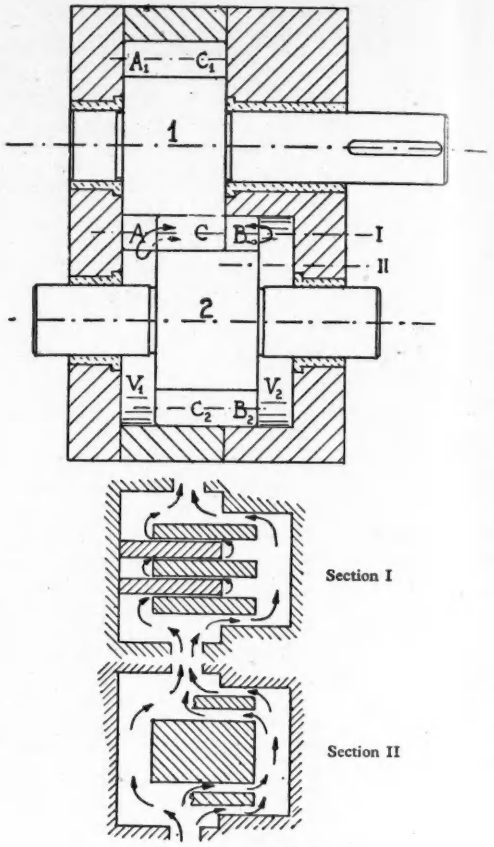


Fig. 2.

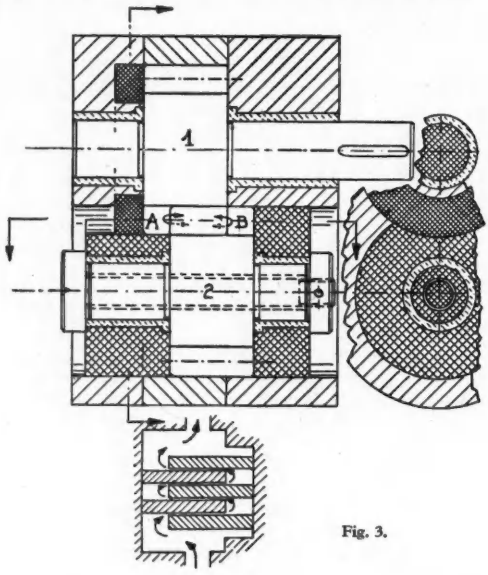


Fig. 3.

small as possible, and the number of teeth as large as possible. A pump of this type will be useful in many cases of minor importance where a coarse continuous control of the discharge during operation is required. The shortcomings are its small volumetric efficiency and its variation in delivery with the delivery pressure ; and these render the pump inadequate in cases where accurate control and close maintenance of the delivery are demanded. These shortcomings are

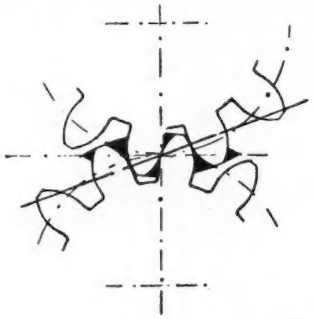


Fig. 4.

particularly inadmissible in a complete hydraulic speed control set of the pump-motor type where the maintenance of constant speed with varying discharge is absolutely essential—and this in combination with a sensitive control.

It should be noted that all pumps of the positive displacement type exhibit a delivery-pressure characteristic which is expressed by the equation for the volumetric efficiency η_v

$$\eta_v = 1 - a\sqrt{\Delta p} - b\Delta p \text{ (Fig. 5)}$$

where the term $\sqrt{\Delta p}$ refers to the reflux losses caused by turbulent flow following the second power law, such as leakages through clearances, gaps, etc. These losses of the turbulent flow variety will be relatively small in a well designed pump. The term Δp however, refers to losses caused by laminar flow, and it is these losses which are preponderant.

The reflux existing in the pump shown in Fig. 3 can be eliminated by cutting the labyrinth path at any point. The only places where these flow paths can be interrupted are the voids *A* and *B*. These voids can be filled either completely or partly in axial direction in the manner indicated in Fig. 6. Here the void *B* is to be occupied by

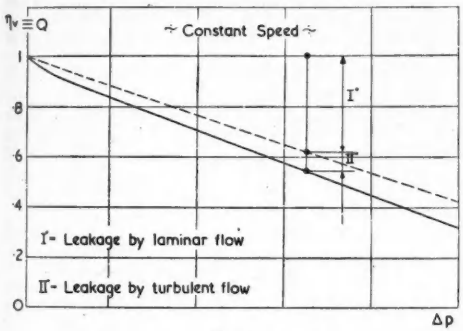


Fig. 5.

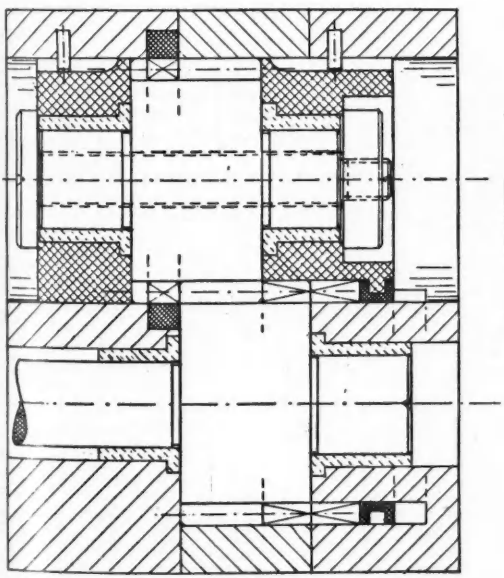


Fig. 6.

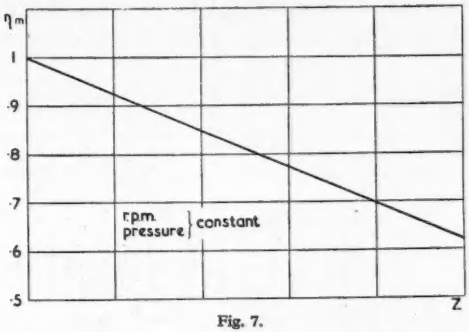


Fig. 7.

an internal gear which rotates with the follower gear but is restrained from moving axially. The elimination of void *A* (Fig. 3) presents a more difficult problem, since it is not possible to employ an internal gear, and a special sealing ring of the type outlined in Fig. 6 must therefore be used.

In the design of a pump of this type attention must be paid to the fact that the number of gear teeth should be a minimum, as the mechanical efficiency of a gear type pump increases with a decreasing number of teeth. The mechanical efficiency, as charted in Fig. 7, is given by :

$$\text{Mechanical efficiency} = 1 - \text{const.} \times \frac{z^2 \times m^2}{z \times m} =$$

$$= 1 - \text{const.} \times z$$

where z is the number of teeth and m is the useful peripheral thrust. In order to keep the mechanical efficiency high, efficient means must be provided to keep the two pistons which constitute the sealing pieces in position, without yielding to the highest axial thrust set up by the internal pump pressure. These requirements call for the provision of adequate thrust bearings, preferably of the ball bearing type, in order to minimize friction. In order to further reduce friction losses, the shafts of the gears may also be carried in ball bearings, as indicated in Fig. 8.

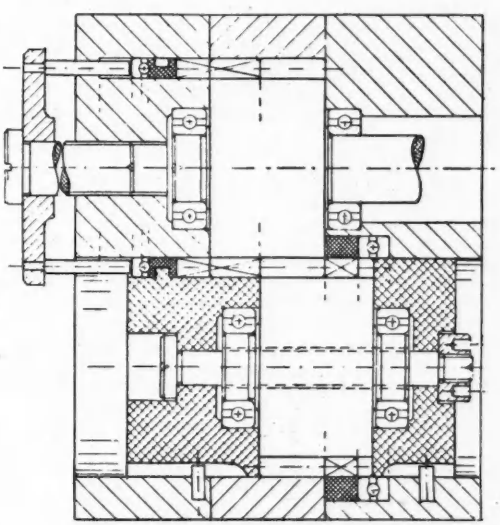


Fig. 8.

RUSSIA

HEAT TRANSFER IN AIR HEATERS.

By V. N. DESHKIN. (From *Sudostroenie* (*Shipbuilding*), No. 1, May-June, 1945 (Published December, 1945), pp. 11-12, 2 illustrations.)

BOILER economy depends to a high degree upon effective heat absorption in the convection surfaces located in the zone of low gas temperature. Accurate knowledge of the heat transfer coefficients obtaining in air preheaters equipped with oval tubing (Fig. 1) is therefore highly

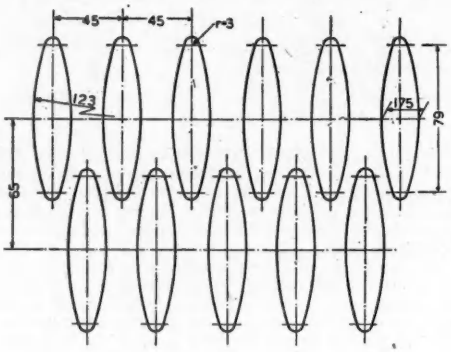


Fig. 1.

important whenever this air preheater type is to be employed. The test results related below were obtained on the air preheater of a boiler fired with mazut.

In these tests the coefficients of overall heat transfer from the gas to the air were obtained, thus permitting a comparison of actual and theoretical values of this coefficient. The ratio of the actual to the theoretical heat transfer coefficient may be expressed by the effectivity factor $\xi = k_{act}/k_{theor}$, where both coefficients are given in cal. per hr. per sq. m. per deg. C. temperature difference. The mean temperature difference between gas and air in cross-flow was established by means of the graph given by Schmidt and reproduced in Fig. 2.

Here $$p = \frac{t_1 - t_2}{t_1 - t_1'}\ ;\ q = \frac{t_2' - t_1'}{t_1 - t_1'}$$

and $\Delta t = r\,(t_1 - t_1')$, where t_1 and t_2 are the gas temperatures at air heater inlet and outlet respectively, and t_1' and t_2' the respective air temperatures. The theoretical heat transfer coefficient k_{theor} is given by

$$k_{theor} = \frac{\alpha_1 \times \alpha_2}{\alpha_1 + \alpha_2},$$ where α_1 is the heat transfer coefficient on the gas side, and α_2 is the heat transfer coefficient on the air side of the heater. Both α_1 and α_2 are given in cal. per hr. per sq. m. per deg. C. According to tests conducted at the Central Boiler and Turbine Institute, the Nusselt number of the air heater under consideration is

$$Nu = 0{\cdot}29\ Re^{0{\cdot}58}, \text{ where } Nu = \frac{\alpha \,.\, d}{\lambda}$$

Here d is the smaller axis of the tube section in metres, and λ is the thermal conductivity of the gas, based on the film temperature. It is also $Re = \dfrac{w \,.\, d \,.\, \rho}{\mu}$, where w is the gas velocity in m/sec, ρ is the density of the gas at the film temperature, given in kg/sec, m^4, and μ is the gas viscosity given in kg/sec, m^2.

The theoretical coefficient of heat transfer on the air side, according to Krausshold, is

$$\alpha = 0{\cdot}024\ w^{0{\cdot}8} \,.\, d_{equ.}{}^{0{\cdot}2}\ \nu^{-0{\cdot}8} \,.\, \lambda \,.\, Pr^{0{\cdot}3}\ \frac{(t_{wall}+273)^{0{\cdot}417}}{(t_{stream}+273)}$$

where w is the average velocity of the air in the air heater, taken as the arithmetical mean of the air velocities at the inlet and the outlet of the heater, $d_{equ.} = 0{\cdot}0201$ is the equivalent tube diameter, ν is the kinematic viscosity of the air, λ is the thermal conductivity of the air, and Pr is the Prandtl number. The factors ν, λ, and Pr refer to the temperature of the tube wall, while the temperature t_{stream} is given by the arithmetical mean of the air inlet and air outlet temperatures. From the computed values listed in the subjoined table, it will be seen that the effectivity factor averages $\xi = 1{\cdot}01 - 0{\cdot}09$. If the tests giving widely divergent values for k are excluded, the average value for ξ will then be found as $\xi_{av} = 1{\cdot}00 \pm 0{\cdot}05$.

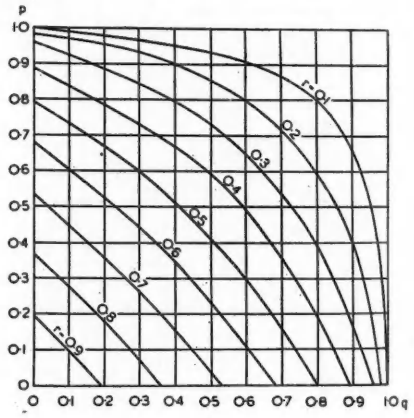


Fig. 2.

Test No.	TEMPERATURES, deg. C. Air Inlet	Air Outlet	Gas Inlet	Gas Outlet	Wall temp. Average	Temper. Diff. "Schmidt" deg. C.	Average air velocity m/sec.	*Re* of gas	*Pr* of air	per cent load	Heat transfer coeff. $k_{theor.}$	$k_{actual.}$	Effectivity factor ξ	$\xi_{av} - \xi$
1	29	207	426	264	232	214	11·5	2940	0·729	52	23·4	25·5	1·09	—0·08
2	31	222	505	315	268	274	15·4	3920	0·733	81	27·8	27·8	1·00	+0·01
3	32	246	576	379	305	321	18·5	4750	0·738	110	31·4	33·9	1·08	—0·07
4	26	205	463	315	252	262	12·4	3770	0·731	60	25·6	23·8	0·93	+0·08
5	28	211	476	333	237	286	13·3	4070	0·730	70	26·3	23·8	0·905	+0·105
6	28	245	559	393	307	328	18·9	4870	0·738	113	32·2	31·4	0·975	+0·035
7	31	205	463	306	248	248	13·1	3660	0·730	51	25·7	25·0	0·97	+0·04
8	38	230	508	348	281	282	16·3	4040	0·735	80·5	28·4	27·6	0·97	+0·04
9	34	220	452	320	261	246	15·3	4080	0·730	80·5	28·5	30·4	1·06	—0·05
10	36	244	515	356	295	283	18·4	4130	0·736	102	30·5	33·6	1·10	—0·09
11	36	204	413	262	217	204	12·0	2880	0·728	53·2	25·8	27·0	1·04	—0·03

SCALE GLASS AS A SUBSTITUTE FOR MICA.

By J. M. STEVELS. (From *Philips Research Reports*, Vol. 1, No. 2, January, 1946, pp. 129-134).

1. PRINCIPLES OF PREPARATION.

IT is a well known fact that two glass plates which are optically flat can cohere so perfectly that they cannot be separated. The same must be possible if the plates are so thin and so flexible that on cohering they are able to follow each others external boundaries. Starting with a large number of glass scales a thick plate may be built up which may be expected to be sufficiently strong and yet flexible enough to fulfil the conditions required for a substitute for mica in radio valves and in other applications.

We have found that such plates can be built up rather easily in the following way : if a large number of tiny glass scales are placed in a suitable medium, the scales are surrounded by an electric double layer. As a consequence the scales repel one another and are kept at some distance apart. The influence of gravity makes them settle and the double layer repulsion effects a parallel orientation of the scales at the end of the sedimentation, so that in the final state all the scales lie parallel to each other. If the medium is now removed, the scales sink together and form the required plate by their mutual van der Waals attraction.

2. REQUIREMENTS FOR THE PRODUCT.

Mica is used in radio valves to fix the electrodes in the required positions and to insulate them from each other. It is very good material for this purpose, since it has a number of favourable mechanical properties. It is sufficiently strong, it can be split and punched very easily, which is of great importance for manufacture. It is flexible and has the property that an electrode stuck into the material is held rather firmly, but nevertheless remains moveable.

The electrical properties of mica as an insulator are also excellent. It has a low electrical conductivity, small dielectric losses ($\tan \delta = 1 \times 10^{-4}$) and a rather high dielectric constant ($\epsilon = 6$). Mica does not decompose below 700 deg. C. Its thermal conductivity parallel to the cleavage plane is 0·011 cal/deg. cm sec, perpendicular to this plane it is 0·001 cal/deg. cm sec. Obviously, the substitute should have about the same properties.

3. GENERAL THEORETICAL REMARKS.

In asking the question, what are the ideal circumstances for the formation of the plates, we must distinguish between two phases :

A. The scales are still gliding in the liquid. In order to prevent flocculation, it is necessary that the interaction potential of the electric double layer surrounding each scale, as a function of the distance, be developed in such a way that, together with the potential of the van der Waals forces, a potential barrier is formed sufficiently large compared with the energy of the Brownian movement of the particles[1]). Generally speaking, this is not the case for liquids with a small dissociating power. Here flocculation with a large sedimentation volume occurs, since the component particles are orientated in all directions. This is the case with glass scales in nitro-benzene, benzene and carbon tetrachloride.

B. If flocculation does not occur, the scales will settle in more or less parallel positions due to their mutual repulsion, so that a very dense arrangement is obtained. These sedimentations are characterized by a very small volume. Since in this case a number of scales are piled up on each other, the potential of gravity begins to play a role. This potential decreases the potential barrier mentioned above, so that the repulsion due to the double layer has to be stronger in order to prevent flocculation. A commercial lead glass (containing about 57 per cent $Si0_2$, 29 per cent $Pb0$, 7 per cent Na_20, 5 per cent K_20 and 2 per cent Al_20_3) gives scales, which do not show flocculation in distilled water or in propanone. Sedimentation, however, takes place rather rapidly, while at many places the parallel arrangement is not realized. If the liquid is now removed a plate of inferior quality is obtained. This disadvantage can be anticipated by giving the scales a sufficient double layer, in other words by giving the liquid a suitable p_H. In the case of the lead glass mentioned above the best orientation of the scales is obtained for $p_H = 8{\cdot}2$ and $p_H = 1{\cdot}2$; for these values plates of the highest quality are obtained. This means that the isoelectric point of the glass lies between these two values, somewhere near $p_H = 4{\cdot}7$.

It is a well known fact that suspensions become stable when their p_H has a value sufficiently different from that at the isoelectric point[2]). In this case, this difference amounts to 3·5 p_H units, which means that the particles of the suspension are stabilized by a double layer of the order of 0·2 volt. The values $p_H = 8{\cdot}2$ and $1{\cdot}2$, respectively are obtained by the addition of a suitable quantity of a solution of NH_3 or HCl. Since in the former case much less electrolyte is present, this method of preparing the plate must be preferred, as the electrolyte has afterwards to be removed completely. The most favourable p_H value for each kind of glass can be determined by stirring the various suspensions. These show the lowest viscosity in their most stable state. In fact, it is found that a suspension of scales of the above-mentioned glass in water shows a minimum of mechanical resistance for $p_H = 8{\cdot}2$ and $p_H = 1{\cdot}2$.

The isoelectric points of most other glasses lie so low that water, with an arbitrary value of p_H between 2 and 12, gives rise to a sufficient double layer. In this case electrolyte-free water must be preferred, and actually the best plates are formed by using distilled water as a medium.

4. INFLUENCE OF THE THICKNESS OF THE SCALES.

The thickness of the scales is of great importance. If they are obtained by blowing out glass bulbs, they will not be quite flat. For this reason they must be so thin, that they can be deformed rather easily by the force of cohesion (possibly combined with the pressure due to their weight), so that they can adhere to other scales over their whole surface. In practice we find that the scales must have a thickness of the order of 1 to 5 μ. Only in this case are compact strong plates obtained.

When starting with scales of about 25 μ a product is obtained with a much looser structure.

5. PRELIMINARY EXPERIMENTS.

Originally the plates were prepared from scales of the above mentioned lead glass. The scales were cleaned by washing with 1 per cent solution of HF, then with concentrated HCl and distilled water. When the suspension is evaporated to dryness, plates are obtained which show coherence and which can be punched. The treatment with HCl is essential ; without it, plates of an inferior quality are obtained[3]).

The preparation was considerably simplified by removing the medium by suction on a suction funnel. In this way several hundreds of different plates were made : during this investigation the following results were obtained :

(*a*) The resulting plates are better, if more suitable media are used. Media with a weak dissociating power give rise to a badly developed double layer. Already in the suspension the van der Waals forces give rise to flocculation (for in-

stance in the case of the above-mentioned lead glass in nitrobenzene, benzene and carbon tetrachloride).

(*b*) For this special lead glass the best results are obtained in water with $p_H = 8{\cdot}2$ or $1{\cdot}2$. As mentioned above, suspensions with these values for p_H are characterized by a minimum viscosity.

(*c*) All other glasses investigated and also quartz are so " negative " that distilled water can be used with success. This has the advantage that, after drying, there are no electrolytic impurities in the plates.

(*d*) The quality of the resulting plate depends on the medium used. The most stable suspension gives the best plates. The sequence for the glass scales investigated is water, methanol, propanone.

(*e*) It is found that, generally speaking, a larger potential difference due to the double layer ($\pm$ 0·2 V) is necessary than in the case of the real suspensions ($\pm$ 0·1—01·2 V) in order to prevent flocculation.

The most favourable electrolyte concentrations will therefore depend on the size, the thickness and the specific density of the scales, the van der Waals forces between the particles and the properties of the medium.

(*f*) The influence of additions to the medium upon the properties of the plates obtained was investigated.

By adding 1 per cent potassium-silicate solution of a specific density 1·3 to the medium, plates are formed which are much stronger, but also more brittle. Bridges between the scales probably come into existence and give rise to these new properties.

Using ethanol as medium 1 per cent ammoniacal phenolaldehyde resin can be added, which produces very hard plates after heating to 100 deg. C.

If propanone is used as medium the methyl ester of polymetacrylic acid can be added in various quantities and the plates obtained are hard and only slightly brittle after drying at 50 deg. C. Obviously, plates obtained in this and in the previous way, cannot be used in radio valves, since they cannot be treated in vacuo to 400 deg. C.

The influence of phosphoric acid was also investigated extensively. By adding 10 per cent of this compound to the medium (in this case water), very flexible, parchment-like plates are obtained, which can even be folded, but which, however, loose their rigidity above 400 deg. C. The most suitable combination of phosphoric acid and potassium silicate for the application in radio valves was sought, and as a result of a systematic investigation it can be stated that the best results are obtained by adding 0·5 per cent of phosphoric acid and 0·2 per cent of potassium silicate to the water.

6. THE PREPARATION OF SCALE GLASS PLATES IN PRACTICE.

After considerating the preceding sections we may give practical directions for the preparation of scale glass plates of good quality from scales of almost all compositions.

The scales can be obtained in large quantities by the following process. The glass is fused and drawn through a slit of 4 mm. width with a length of about 100 mm. At both ends the slit ends in a circular hole with a diameter of about 10 mm. The glass drawn from this dumb-bell shaped slit emerges as a membrane suspended between two rods. When the glass is cooled, the rods are removed by cutting. The remaining membrane is very easily broken up into small particles which are used as raw material for the production of the scale glass.

Glass scales with a thickness of about 1 μ and a surface of about 1 mm^2 are placed in distilled water to which a little phosphoric acid and potassium silicate (according to section 5f) is added. The scales are given some time to settle in the suction funnel before the suction is applied. A wet slimy plate results. The settling of this plate can be helped by rubbing it lightly.

When the plate is sucked thoroughly dry, it is still deformable : it can now be given any arbitrary form. After drying the resulting plate at 80 deg. C. for an hour, it remains fixed in its ultimate form.

For use in radio valves, drying at 80 deg. C. is not sufficient. Upon evacuating the valve a quantity of enclosed water escapes from the plate and this may give rise to an explosive shattering of the plate. For this purpose it is necessary to heat to 200 deg. C., if possible under light pressure. For use in radio valves it is advantageous to cover the plates with a thin layer of SiO_2 by spraying with a solution of ethyl silicate in a water-methanol mixture.

Following the method given, it is possible to produce scale-glass plates of hard glass (containing only SiO_2, B_2O_3 and Al_2O_3) which can compete in all respects with mica in radio valves. Valves equipped with plates of 0·5 mm. could be subjected to shock for 6 hours and could be dropped on the floor without fracturing.

Hard glass is used by preference, since otherwise the glass may fuse slightly around the cathode during its formation.

7. COMPARISON OF THE PROPERTIES OF MICA AND SCALE GLASS.

The properties of mica and scale glass are compared in the following table :

		Mica	Scale glass
	Mechanical properties.		
	Cleavability	+	—, but any desired thickness > 30 μ can be made [1]
	Permits punching	+	+
	Flexibility	+	+
	Smoothness [2]	+	+
	Deformability [3]	—	+
II	*Electrical properties.*		
	Electrical conductivity the same for mica and hard glass.		
	Dielectric losses	1×10^{-4}	5×10^{-4}—40×10^{-4} [4]
	Dielectric constant	6	3—6 [4]
III	*Thermal properties.*		
	Thermal durability	to 700 deg. C.	to 1000 deg. C. (for hard glass)
	Thermal conductivity	// cleavage plane 0·011 cal/deg cm sec	// surface 0·002 cal/deg cm sec [5]
		$\perp$ cleavage plane 0·001 cal/deg cm sec	$\perp$ surface 0·0003 cal/deg cm sec [5]

Explanations of the table.

[1] Plates up to 1 mm thickness were made, but this is certainly not the limit.

[2] This is the property which means that a pin can be stuck through the material and held rather firmly but nevertheless remains removable (of importance in case of application in radio valves).

[3] This is the property which indicates that wet plates may still be shaped to a desired form which becomes fixed after drying.

[4] These properties are dependent on the glass used. Generally speaking, the dielectric losses of scale glass of a certain composition are the same as for the corresponding compact glass, whereas the dielectric constant is about half the value of that of the corresponding compact glass.

[5] These data are measured for the lead glass mentioned in section 3.

8. APPLICATIONS.

Scale glass can be used with success in the following cases :

(a) *In radio valves.*

Here scale glass can be substituted for mica. The emission of the valve is not affected by the substitution of scale glass for mica. Advantages of the former are

higher temperature resistance, lower thermal conductivity. The property of deformability gives the possibility of producing bent plates.

(b) *Transmission valves.*

Here the dielectric losses of the usual glasses are a drawback. Recently developed glasses, however, (with $\tan \delta = \pm 5 \times 10^{-4}$), which have, up to now, not been tried as scale glass, may offer a solution.

(c) *Other applications.*

For heating elements in flat irons, electric kettles and other domestic apparatus, scale glass is a suitable material, as was proved by experiments with constant load. The property of deformability is here of great advantage compared with mica. The possibility of producing plates of varying thickness, generally speaking, of three-dimensional construction is of great importance. Also the possibility of " baking in" wires etc., may be of importance.

(d) *Thermal insulation.*

Finally it may be remarked that scale glass is a good thermal insulator. In this case, scales of about 25 μ thickness must be taken, so that a loose structure is obtained.

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RUSSIA

CENTRIFUGAL CASTING IN THE SHIPBUILDING INDUSTRY.

By N. N. SOCOLOV. (From *Sudostroenye*, No. 3/4, 1945, pp. 25-31, 12 illustrations.)

AT a Russian foundry a centrifugal casting machine is employed for the casting, in metal moulds, of bronze ships propellers of up to 700 mm. (27⅝ in.) diameter. As will be seen from Fig. 1, the moulds (2) are placed on the turntable of the verticle spindle (1) of the casting machine, the latter being enclosed by the circular guard (3) and the sheet metal housing (4).

The metal mould consists of a cast iron base, and of three cast iron cover pieces, one of which is shown in Fig. 2. At the top, the three cover pieces are bolted to a common flange. The component parts of the mould are cast from patterns. The working surfaces of the mould are machined by a method similar to that employed in the fabrication of propeller blades. The correct relative positions of the various parts of the metal mould are insured by locating pins. In order to ensure proper rotational balance, the assembled mould is statically balanced.

The seepage of liquid metal through the joints of the mould is prevented by grooving the abutting faces of the individual parts and inserting asbestos cord. Prior to casting, the mould surfaces are greased with a paraffinic mazut. After that, the top parts are placed upon the base and attached to the latter in the manner shown in Fig. 1. The flange carrying the pouring funnel is then bolted to the upper part of the mould.

In order to overcome the considerable chilling effect of the mould surfaces, the mould is preheated to 450-500 deg. C. (842-932 deg. F.) before it is placed upon the turn table of the casting machine. The funnel is also preheated to some 250-350 deg. C. (482-662 deg. F.).

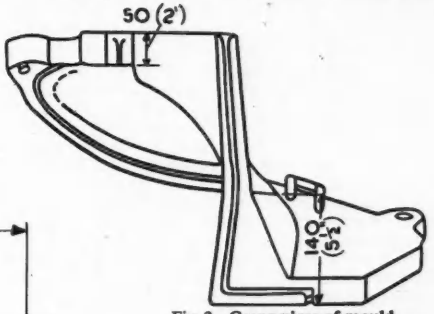


Fig. 2. **Cover piece of mould.**

The metal is poured at a temperature of 1020-1040 deg. C. (1868-1904 deg. F.). The rotational speeds employed for casting range from 225 to 335 r.p.m. The castings are removed from the mould one hour after pouring. The following advantages are claimed for this method of casting bronze propellers : (1) Due to the decrease in blade thickness achieved, ship propellers of the size indicated in the accompanying illustration weigh only 110 kg. (243 lb.) as compared to 320 kg. (706 lb.) in the case of propellers cast by the ordinary method. (2) the employment of a metal mould affords a saving of 85 man-hours in the casting of each propeller. (3) the number of defective castings is greatly reduced. (4) the reduction in the thickness of the blades results in a saving of 100 man-hours in the machining of the casting.

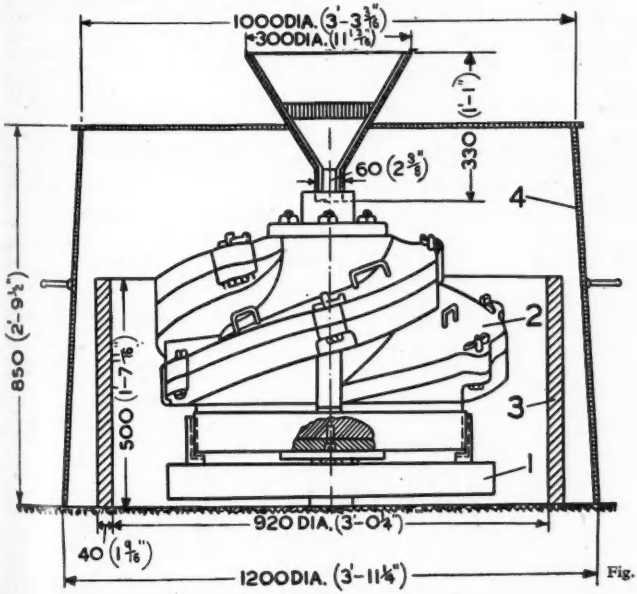


Fig. 1. **View of machine for centrifugal casting of ships' propellers.**

INVESTIGATION OF ROLL WEAR IN THE HOT ROLLING OF WIRE AND STRIP.

By R. TOSTERUD AND E. STENFORS. (From *Jernkontorets Annaler*, Vol. 130, No. 6, June, 1946, pp. 213-238, 28 illustrations.)

THE purpose of the investigation referred to was to establish methods for the continuous routine control of roll wear in the hot rolling of wire and strip, and also to derive the influence of certain factors upon roll wear from the test results obtained.

In rolling mill operation the roll grooves must be of perfect form and surface quality ; any defects in this respect will at once be revealed by a faulty product which in certain serious cases will have to be rejected.

Faulty shape—that is, " macro-stereometric defects " —is a direct consequence of normal wear. In the rolling of quality wire with narrow diameter tolerance the rolls can be used only as long as they allow the rolling of wire within the permitted final wire diameter tolerance. In the case of wire of lower quality the tolerances are usually so large that the grooves can be worn down to a point where the roll surfaces begin to come into contact, provided that the permitted ovality of the finished wire is not exceeded.

Faulty surface, which is to say " micro-stereometric defects " of the roll grooves, are due to the use of unsuitable roll material as well as to inappropriate rolling conditions.

The maximum radial wear of the roll groove can serve as a measure of the magnitude of faulty shape. The maximum depth of the groove is measured before and after rolling. Measurement was carried out by means of a dial gauge having a measuring range of 10 mm. with an accuracy of 10 microns. The following method was used in carrying out measurements of the depth of the grooves. The dial gauge was placed above the groove in such a way that the gauge feeler coincided with the roll radius. By shifting the gauge holder in axial direction the deepest point of the groove could be found. At first the device shown in Fig. 1, was used, which gave sufficiently accurate results where large curvature radii were concerned. But in the case of small radii this method proved unsatisfactory, and the device illustrated in Fig. 2 was therefore developed. Here the dial gauge is fixed in a holder attached to a sliding carrier moved by means of a screw spindle in longitudinal direction. With this device the groove depth was obtained as the difference between readings for the deepest point of the groove and the average of the readings for the adjacent roll surfaces. Correction for wear of the roll surface was made.

In order to gain an idea of the distribution of the wear over the groove section, lead alloy, plaster of paris, and sulphur impressions were taken. But as faultless grinding of the cross sections of such impressions proved

Fig. 1. **Holder with depth gauge for measuring the groove depth of wire mills.**

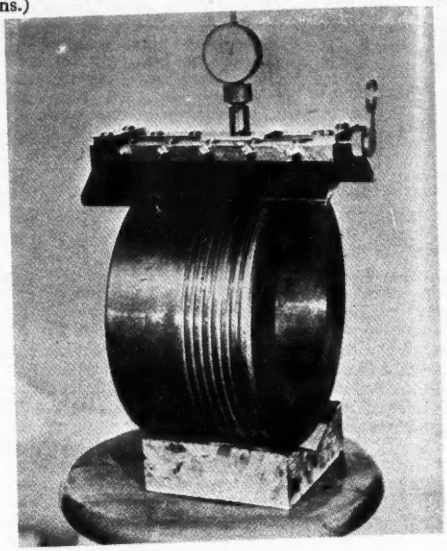


Fig. 2. **Device for measuring the groove depth of the rolls of wire mills.**

highly difficult, this method was abandoned, and an attempt was made to produce shadow photographs of the plate-shaped lead impressions secured by pressing a piece of lead (*e*) into the groove by means of a punch (*g*), as indicated in Fig. 3. From the impression obtained, a shadow photograph was then made with the use of a profile projector. In order to secure sharp pictures the impressions must be as thin and plane as possible. By obtaining such photographs before and after rolling, the difference in groove contour due to wear was then found as shown in Fig. 4, where the shaded portion represents the wear suffered by the groove. The easiest and quickest method, however, proved to be the light cross section method according to G. Schmaltz.

The wear of rolls for strip rolling is conveniently determined from the difference in roll diameter before and after rolling. Since roll wear is bound up with the coefficient of friction between roll and piece, it may be of interest to study the roughness of the roll groove, this being of influence upon the magnitude of the friction coefficient. As a measure of roughness in present day practice the " profile depth " H_R introduced by G. Schmaltz (Fig. 5) is used. Both mechanical and optical devices of satisfactory accuracy are available, but for the present investigation the Schmaltz profile micro-

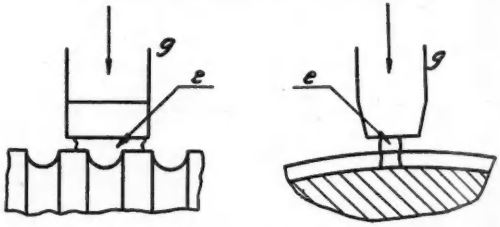


Fig. 3. **Method of taking a lead impression of the roll groove.**

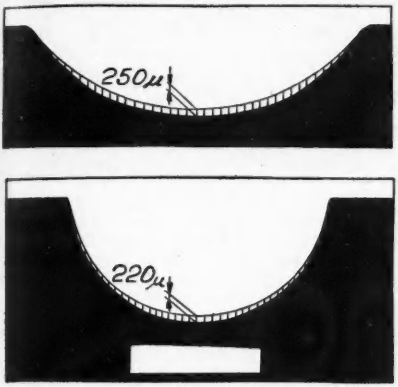


Fig. 4. Top : **Shadow photograph of oval groove before and after use.**
Bottom : **Shadow photograph of round groove before and after use.**

scope was preferred because of its special suitability for this particular purpose.

The factors which influence roll wear are :
(1) Roll material and its structure ; (2) Roll diameter ; (3) Groove shape ; (4) Analysis of the material rolled ; (5) Quantity of material rolled ; (6) Dimensions of the piece rolled both before and after rolling ; (7) Rolling temperature ; (8) Rolling speed ; (9) Water-cooling during rolling.

In the tests conducted, the influence of the roll material and its structure, of the groove shape, and of the analysis and the quantity of the material, were studied, as well as the influence of the rolling speed and of the initial height of the piece. The roll diameter, the percentage linear pressure, the water cooling (which was ample), and with it the friction coefficient between piece and roll, were held as constant as possible, since the possibility of varying these was small in the case of the continuous wire mill which was used for the majority of tests.

In order to be able to compare the magnitude of roll wear under the various test conditions, the concepts of " wear quotient " and " relative wear " had to be introduced. This quotient, which it is convenient to use in practice, indicates the ratio between maximum radial roll wear and the corresponding amount of material rolled in microns per ton ; while the relative wear is represented by the ratio of the maximum radial roll wear and the total length of material rolled in microns per kilometre. The phenomena taking place in the roll gap must be discussed in detail in order to explain these concepts. A theoretical analysis of the influence of the rolling factors upon the roll wear can hardly be given, since the complex phenomena taking place in the roll gap are not fully clarified as yet. However, by resorting to a considerably simplified concept, the influence of some of the various factors upon roll wear can be estimated.

Referring to Fig. 6, it will be seen that at point B the material has a lower speed than the roll, and at point M

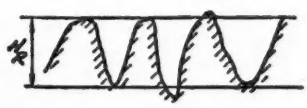


Fig. 5. **Profile showing surface roughness.**

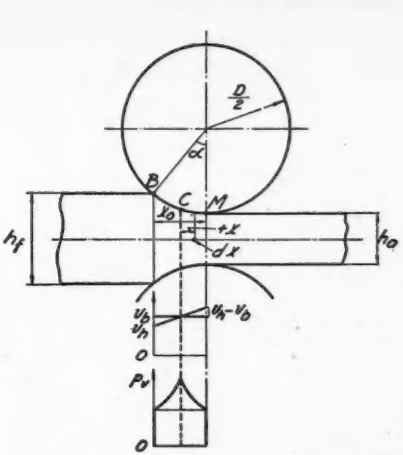


Fig. 6. **Diagrammatic representation of the phenomena taking place in roll gap.**

a higher speed. Between these two points there will exist a point C at which the speeds of material and roll are equal. Within the intervals BC and CM slipping of the material relative to the roll will take place in opposite directions ; and it is this slipping which causes roll wear.

The important feature in the mechanism of roll wear may be assumed to be the slipping under pressure, and roll wear can therefore be conceived as originating in part from the oxidation of the roll surface where it contacts the material being rolled. One would assume the existence of a linear relationship between the work of friction performed and the amount of material worn off per unit surface area of the roll. This assumption is confirmed by experiments which show that the wear of one and the same groove is in proportion with the length of material rolled, provided all other rolling factors are kept constant.

As the amount of material worn off per unit area of roll surface can be expressed by the maximum radial roll wear in microns, the roll wear ΔS per roll revolution will be given by $\Delta S = k_1 \cdot A$, where ΔS is the maximum radial roll wear per roll revolution, k_1 is a constant, and A is the work of friction performed per mm roll width and revolution.

Assuming the percentage linear pressure, the roll diameter, the friction coefficient, and the groove contour to remain constant, the radial roll wear after N revolutions can approximately be expressed by

$$S = k_2 N p_{r\,av} \sqrt{h_a}$$

where k_2 is a constant, $p_{r\,av}$ is the average specific roll pressure, and h_a is the height of the material after the pass. The pressure $p_{r\,av}$ may vary considerably with the rolling factors. In the case of wire rolling where the sections of the material are of similar shape, the wear figure s is defined as :

$$s = \frac{S}{\Sigma G} = \frac{k_2 N p_{r\,av} \sqrt{d_a}}{\gamma \Sigma l \dfrac{\pi d_a^2}{4}}$$

where γ is the specific gravity of the material and d_a the diameter of the wire after the pass. It is therefore :

$$s = k_3 p_{r\,av} d_a^{-1.5}$$

where k_3 is a constant.

The chill cast rolls of usual standard quality had the following analysis :

Rolls for Wire Rolling :

3·2 — 3·4	per cent	C
0·4 — 0·7	„ „	Si
0·3 — 0·5	„ „	Mn
< 0·4	„ „	P
< 0·15	„ „	S

Rolls for Strip Rolling :

3·5 — 3·7	per cent	C
0·5 — 0·8	„ „	Si
0·2 — 0·3	„ „	Mn
< 0·4	„ „	P
< 0·3	„ „	S
0·15	„ „	Cr

Some of the grooved rolls for wire rolling were alloyed with 0·5 per cent Mo, but as the molybdenum content was found to be without influence upon the service life of the rolls, no differentiation between rolls alloyed and those not alloyed with molybdenum will be made here. The hardness of the 30-35 mm case was 46 to 48 Rockwell. Three types of material were rolled, namely :

I. Soft commercial iron with 0·1-0·15 per cent C ;
II. Valve spring steel with approx. 0·6 per cent C ;
III. Ball bearing steel with 1·0-1·10 per cent C and 0·2-0·4 per cent Cr.

The final rolling temperatures for wire varied within such narrow limits that they could be taken as practically constant at 910 deg. C., representing the mean of the initial temperature t_1 and the final temperature t_2 of the heat rolled. The temperature drop $(t_1 - t_2)$ was practically constant at 25 deg. C. The final rolling speed as a function of the final dimensions is given in Fig. 7.

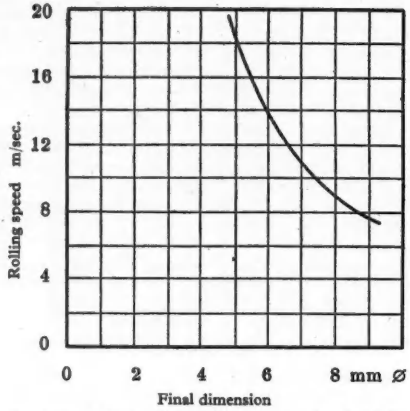


Fig. 7. **Approximate relationship between final rolling speed and final dimension.**

The reduction is difficult to determine under operating conditions, but can generally be taken as approx. 10 per cent for all dimensions in the last pass. In the case of strip rolling the speed of the strip after the last pass varied between 5·0 and 5·6 m per sec., while the final rolling temperature fluctuated between 785 and 840 deg. C. The temperature drop of the heat from commencement to termination of rolling varied between 30 and 15 deg C. according to the weight of the coil and the final dimensions. In order to facilitate evaluation of the test results, these variations in temperature and in rolling speed were neglected.

Fig. 4 characterizes well the type of shape defects arising as a consequence of ordinary wear. The upper part of this illustration shows the finished oval contour and the lower part the finished round contour for the rolling of 9·5 mm. wire of material No. III. In the case of the oval groove 37·4 tons of wire were rolled, and the linear wear is seen to be evenly distributed over almost the entire width of the groove. In the round groove 40·8 tons of material were rolled, and the wear is seen to be greatest at the centre of the groove, thus justifying the practice of taking the wear at the centre as a measure of contour defects. The lesser wear on the sides of the groove is due to the lower surface pressure and partly also to smaller length of contact between material and roll in these zones. It will also be seen from the illustration that the wear is noticeably greater in the left hand part of the groove. This may be taken to be due to inaccurate entering of the opposite grooves of the rolls.

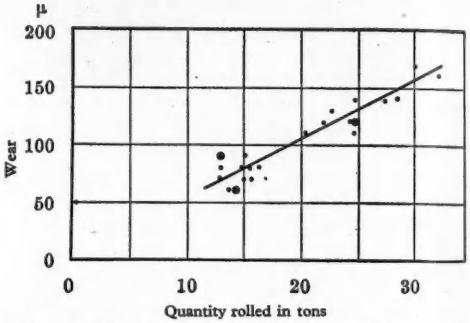


Fig. 8. **Relationship between linear wear of groove and quantity of material rolled. (4·9 mm. wire of material No. I).**

The relationship between linear groove wear and the quantity of 4·9 mm. wire rolled is charted in Fig. 8, which refers to the rolling of the soft material No. I. The corresponding relationship for the hard valve spring wire (material No. II) is given in Fig. 9. It can be concluded that the friction coefficient remained constant during the different stages of the use of the groove, the originally smooth groove becoming roughened when rolling the first few rods. After that the surface roughness remains almost constant.

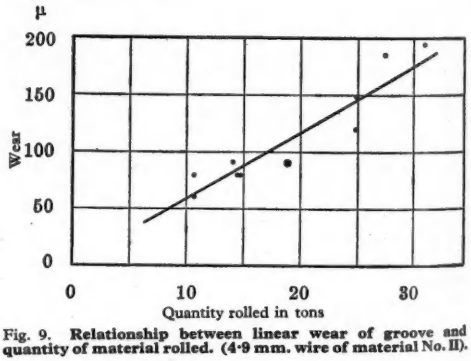


Fig. 9. **Relationship between linear wear of groove and quantity of material rolled. (4·9 mm. wire of material No. II).**

The results of the investigation may be found useful when planning a rolling mill for a given output or when buying new rolls, as they allow an approximate estimate to be made of the rate of roll replacement to be expected. Furthermore, the resistance to wear of different roll materials can be compared by this method.

THERMAL DE-GASSING OF BOILER FEED-WATER.

By J. TEYSSLER. (From *Strojnicky Obzor*, Vol. 26, No. 1 and 2, 1946, pp. 9-12 and 20-25, 28 illustrations.)

INTRODUCTION.

CORROSION of metals in steam-raising plants is due to chemical or electro-chemical reactions in which water takes the place of the electrolyte.

There are many methods of combating corrosion in boilers, pipe lines, etc., of which the removal of oxygen from the feed-water is of prime importance.

If oxygen is concentrated locally, in the form of a gas bubble on the wall, it forms a galvanic cell with the metal of the wall. Oxygen helps the corrosion by depolarizing the galvanic cell.

Ferrous hydroxide is produced by hydrolysis :—

$$H_2O = H^{\cdot} + (OH)'$$

$$Fe^{\cdot\cdot} + 2\,(OH)' = Fe\,(OH)_2$$

and this is further oxidized as follows :—

$$4Fe\,(OH)_2 + O_2 = 2Fe_2O_3 + 4H_2O \quad .. \quad (1)$$

or $$2Fe\,(OH)_2 = 2FeO + 2H_2O$$

and $$4FeO + O_2 + xH_2O = 2Fe_2O_3 + xH_2O \quad .. \quad (2)$$

The action of oxygen is further increased by the presence of free carbonic acid :—

$$Fe_2 + H_2CO_3 = Fe_2CO_3 + H_2 \quad .. \quad (3)$$

Fe_2CO_3 is an insoluble salt which forms a protective film on the surface of the metal. In the presence of oxygen, however, rust develops and carbon dioxide is evolved :—

$$2Fe_2CO_3 + 2H_2 + 3O_2 = 2Fe_2O_3 + 2H_2O + 2CO_2 \quad (4)$$

so that reaction (3) continues without interruption.

From the above it is obvious that the primary aim of de-gassing feed-water is the removal of oxygen.

This requirement must also be satisfied when using condensate as feed-water and it is not sufficient to de-gass the make-up only, as it is often done. Although the condensate from turbines is free from gases, leakages in the pipe lines and pumps allow atmospheric oxygen to enter into the system where it is readily absorbed. Experiments made with condensate at rest, conducted by Splittgerber[12*] show (Fig. 1) the rate of absorption of the oxygen.

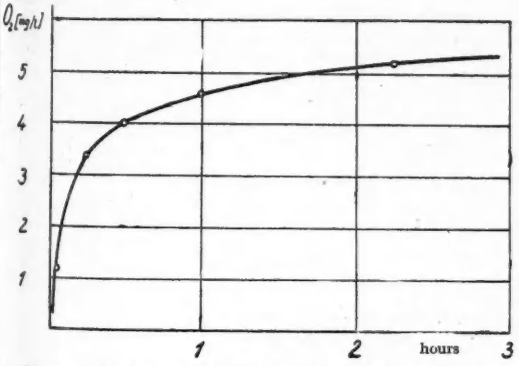


Fig. 1. **Rate of absorption of oxygen from air by de-gassed condensate.**

Chemical de-gassing is achieved by the binding of oxygen dissolved in the water by means of chemicals such as sulphur dioxide and sodium sulphite[3]. The drawback of this method lies in the fact that the water contains impurities from the residues of the chemical process. The last few decades lead therefore to a general acceptance of thermal de-gassing. By this method not only oxygen is removed but also other gases including

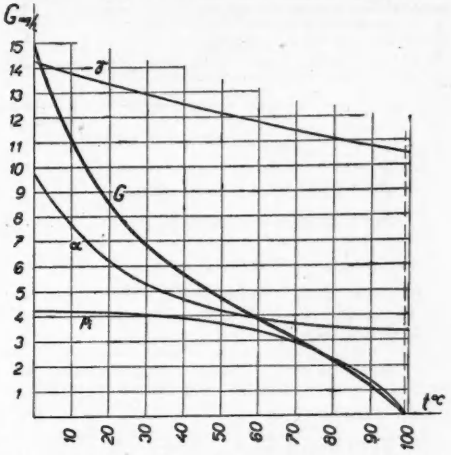


Fig. 2. **Weight of oxygen in water as a function of temperature at a pressure of 760 mm Hg, according to equation** $G = \alpha\,\gamma\,p_1$

carbon dioxide and chlorine which are often present in water coming from aqueducts.

THE PRINCIPLE OF THERMAL DE-GASSING.

The absorption of gas in a liquid is directly proportional to the pressure of the gas. The absorption of a mixture of gases takes place in proportion to the partial pressures of the constituents of the mixture.

Generally $V_1 = \alpha_1 p_1$

where V_1 is the volume of gas absorbed, α_1 the absorption coefficient, and p_1 the partial pressure.

In order to reduce the volume V_1, it is necessary to reduce α_1 or p_1, or both. By increasing the temperature of the feed-water, the absorption coefficient decreases, and the pressure of the water vapour increases. At the boiling point of the water, corresponding to the pressure p, the space above the water level is filled with saturated steam of partial pressure p, whereas the partial pressure of the other gases is zero ; that is to say, the liquid is

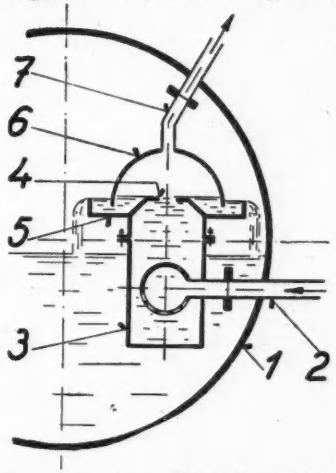


Fig. 3. **Pressure de-gassing within the boiler (Hüllsmeyer-Düsseldorf).**

*Numerals refer to the bibliography at the end of this article.

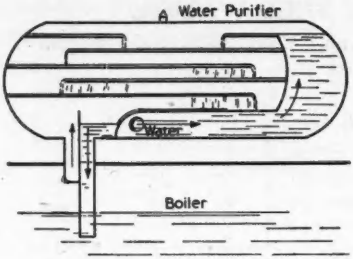


Fig. 4. **Locomotive purifier.**

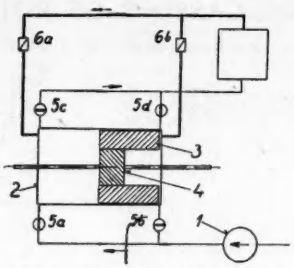


Fig. 5. **De-gassing in the delivery branch of the feed pump.**

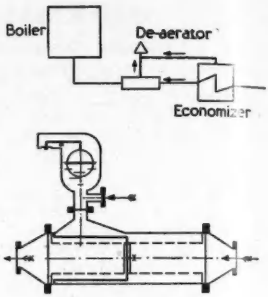


Fig. 6. **De-gassing in the economizer.**

free of gases. This is shown in Fig. 2 for oxygen, where γ is the weight of oxygen per unit volume, and G its weight in mg/litre.

The practical application of this principle to feed-water de-gassing can be divided into two groups, that of vacuum de-gassing and that of pressure de-gassing.

Vacuum De-Gassing.

This can take place at normal or at elevated temperatures according to the required quality of the feed-water. It can be carried out in the surface condenser of a turbine or in suitably equipped water tanks. In the former case the efficiency of the condenser is reduced, in the latter case a separate vacuum pump is required.

Special precautions are necessary to prevent air from being drawn into the system ; for this reason vacuum de-gassing has been abandoned in recent years in favour of pressure de-gassing. For further information on vacuum de-gassing see bibliography at the end of this article.[4, 5]

Pressure De-Gassing.

When de-gassing at a pressure higher than atmospheric, it is necessary to heat the water to boiling point in order to remove completely the dissolved gasses. De-aerators, fitted into the feed line, only remove the air freely separating from the water ; but they do not achieve complete de-gassing of the feed-water.

(a)—*De-gassing at high pressure.*

Fig. 3 shows an example of an apparatus for de-gassing water within the boiler itself. The water is led through pipe 2 into a prismatic vessel 3, where it is pre-heated. From there it flows through openings 4 into trough 5 before entering the boiler itself. At the outlet from openings 4, the water is under somewhat lower pressure ; the gases escape into collector 6 and are led away through pipe 7.

The common water purifier, as in use on locomotives, is shown in Fig. 4. Apart from removing the temporary hardness of water (by de-composition of acid carbonates with increasing temperature) it is a very efficient spray de-gassing equipment. An ingenious method of de-gassing is shown schematically in Fig. 5. Water from the feed pump 1 enters cylinder 2, in which there is a piston 3, containing in an annular space piston 4. The valves 5 are controlled by piston 4. When 5a and 5d are open 5b and 5c are closed and *vice versa*. The water entering cylinder 2 *via* valve 5a, drives piston 3 to the right hand side and discharges the water from that side *via* valve 5d into the boiler. Piston 3 carries along with itself piston 4 ; when reaching the right hand end position piston 3 stops (it is driven by water pressure only), 5a and 5d close, and piston 4 moves further to the right since it has a separate drive. Thereby the pressure drops in the left-hand side of cylinder 2, return valve 6a opens, and steam heats the water in the cylinder to a temperature corresponding to the pressure. By opening 5b and 5c, the hot, gas-free water is admitted to the boiler.

Fig. 6 shows de-gassing carried out in the economizer before the water enters the boiler. (This arrangement is becoming obsolete).

(b)—*De-gassing at low pressure.*

In a cascade de-gassing plant water is heated by steam passing through a nest of tubes. The rising gases are exhausted (or in the case of low pressure, sucked away by a vacuum pump). The steam generated is condensed by the in-flowing cold water which is thereby heated[6].

In the arrangement shown in Fig. 7, a tube connects the heater O to the de-gassing vessel N.

Another installation is shown in Fig. 8. Heating is obtained in a very effective way by leading the mixture of water and steam through tubes. The mixture impinges upon a roof-shaped sheet and very complete de-gassing is thus achieved.

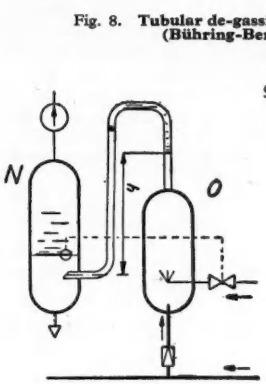
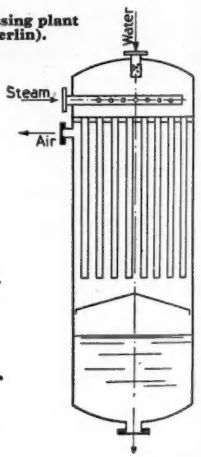


Fig. 7. **Dual de-gassing plant (Kraftanlagen A.G., Heidelberg).**

Fig. 8. **Tubular de-gassing plant (Bühring-Berlin).**



The next two types to be described are intended mainly for use in ships.

In Fig. 9, vessel 3 is equipped with three de-gassing chambers 4. Pump 1 pumps water into the distributor 2 wherefrom it is led into the chambers 4 by pipes 5. In each of the chambers 4 there are over-flow tubes 6, connecting the spaces 4 and 3. The total cross-sectional area of tubes 6 equals that of tubes 5. The water in the de-gassing chambers 6 is heated by steam through pipes 7. Due to the different heights of the horizontal branches of tubes 5, assume that, at small loads, the right chamber 4 only obtains water from 2.

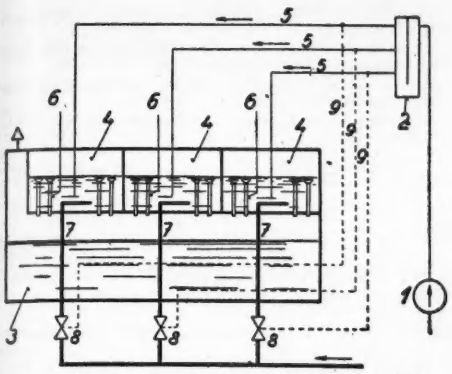


Fig. 9. **Marine de-gassing plant (Deutsche Schiff-u. Maschinenbau A.G.—Bremen).**

By remote control 9, valves 8 interrupt the steam supply to the remaining two chambers.

In Fig. 10, water in a closed vessel is heated by low-pressure steam and passes through over-flow tubes into a container. The over-flow tubes can be of different design. Arrangement *A* is stationary, *B* rotating about its own axis. Water, moving down the tube is forced to the outer periphery by the action of centrifugal force, while the substantially lighter gases can escape through holes at the tubes' inner periphery. The pressure (as indicated in Fig. 10) is chosen so as to enable most complete de-gassing.

Fairly generally in use are the spray de-gassing plants. In these water is distributed over a system of perforated sheets (sprays) leaving just enough passages for the escaping gases (Fig. 11). In Fig. 12 the perforated sheet at the bottom of the containers is sloping on both sides so that the water flows in form of cascades, an effect that could also be obtained in the arrangement of Fig. 11 by suitable perforation. Fig. 13 shows a plant in which, for better heating, water flows through a number of small troughs.

The paper describes briefly two spray de-gassing plants handling 60 tons/hr. and 96 tons/hr. respectively, built by the Elliott Co., Jeanette, Pa., and two vacuum

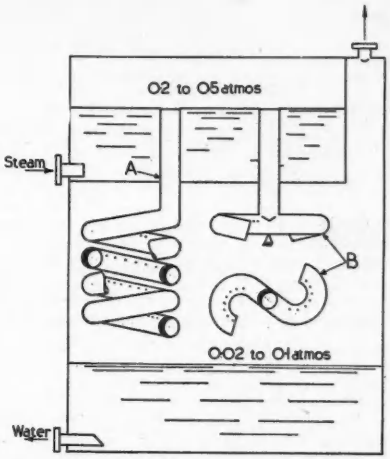


Fig. 10. **Marine de-gassing plant (Deutsche Schiff-u. Maschinenbau A.G.—Bremen).**

de-gassing plants handling 270 tons/hr. and 450 tons/hr. of the Cochrane Corp., Philadelphia, Pa. These are well-known to the English-speaking engineer and therefore this part of the paper is omitted.

A brief description of the problems concerning the feed pump does not bring any details beyond those well known to the specialist in this field of engineering. The reader is referred to ref.[9] for further information.

The author ends his paper by dealing with the various methods of sampling and examination of water. For the analysis of oxygen, the method of Winkler[10, 11], the method of Alsterberg[10], and the method of Rideal-Stewart-Winkler[12] are described.

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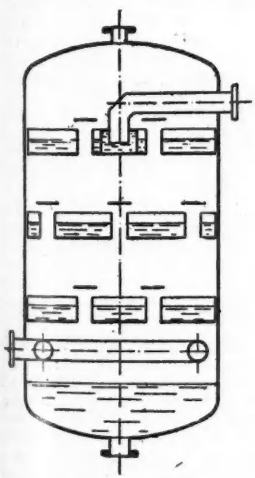

Fig. 11. **Spray de-gassing vessel with bottom made of perforated sheet.**

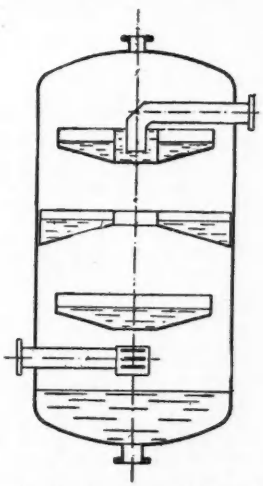

Fig. 12. **Spray de-gassing vessel with bottom made of perforated sheet.**

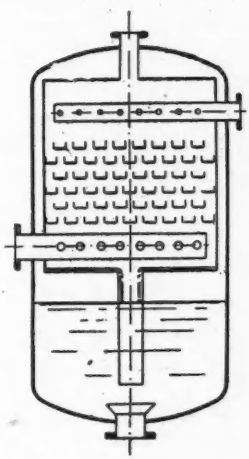

Fig. 13. **Spray de-gassing plant.**

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RUSSIA

HYDRAULIC ACCUMULATOR FOR CONSTANT PRESSURE OPERATION.

By J. KURLOVITS AND F. SVINDENKOV. (From *Zheleznodorozhni Transport*, No. 4, 1946, pp. 84-85, 2 illustrations.)

THE employment of hydraulic accumulators for variable pressure operation for the purposes of water supply in railroad service as well as for the operation of hydraulic presses in industry is resorted to solely for the reason that a simple constant pressure type has not hitherto been available. The authors have succeeded in developing a gas-filled hydraulic accumulator for constant pressure operation in which the required pressure is created not by compressed air but by a gas which evaporates at the temperature and pressure prevailing in the hydraulic accumulator.

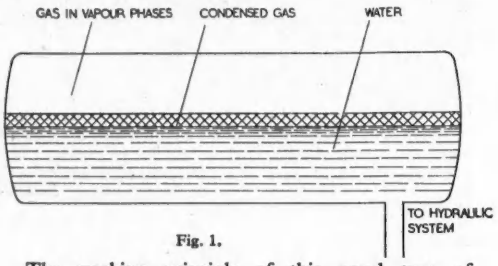

Fig. 1.

The working principle of this novel type of accumulator is outlined in Fig. 1. Here it will be seen that the required hydraulic pressure is created by the generation of gaseous vapours within the accumulator vessel. For any variation in the water contents of the accumulator the prevailing pressure will be governed solely by the temperature. However, the latter remains constant during the operation of the accumulator, and the accumulator pressure will therefore also remain constant. Owing to the slow rate at which the periodic expansion and compression of the gas proceeds during operation of the accumulator, sufficient time is available for temperature equalization of the accumulator contents with the circumambience to take place.

Accumulators for railway water supply service may be of the kind indicated in Fig. 1. In this case the accumulator vessel is preferably buried underground in in order to prevent freezing of the water contents and to keep the latter at a constant temperature. Automatic valves for water level control should be installed.

A mixture of methane and propane serves as working gas, the proportion of the mixture to be chosen in accordance with the pressure to be maintained. In the following example the quantity of butane-propane mixture required for a water accumulator of 200 cu.m. content for a working pressure of 3 atm. is computed.

On the basis of a working temperature of 6-10 deg. C. a mixture of 50 per cent butane and 50 per cent propane will be chosen. Under the given pressure and temperature conditions the density of the gas mixture will be $\frac{(2{\cdot}46 + 1{\cdot}87)}{2} \times 3 = 6{\cdot}48$ kg. per cu.m. and the specific gravity of the condensed gas will be

$$(0{\cdot}582 + 0{\cdot}509)/2 = 0{\cdot}545 \text{ kg. per litre.}$$

In order to obtain one cu.m. of gas $6{\cdot}48/0{\cdot}545 = 11{\cdot}8 \approx 12$ litres of condensed gas mixture will be required. For a working volume of 200 cu.m. the required quantity of condensed gas is therefore found as $12 \times 200 = 2400$ litres by volume, or $2400 \times 0{\cdot}545 = 1310$ kg. by weight.

In view of the low specific gravities of both liquid butane and propane, the mixture will always float upon the water surface. Moreover, no mixing of the butane-propane mixture with the water will occur ; nor will the mixture react with the water contents of the accumulator.

The useful working volume of an accumulator of this type will be high ; theoretically it will amount to 98·8 per cent of the entire accumulator volume. If allowance is made for the small leakage which will occur, the useful working volume may be taken as 95 per cent. This means that practically the entire cubical contents of the accumulator represent useful working volume, which is quite impossible to attain with an ordinary accumulator. Additional advantages of the gas-filled accumulator as compared with the ordinary accumulator type are :

(1) Practically constant pressure.
(2) Simplicity and reliability.
(3) Some 2-2·5 times less steel per cu.m. of volume are required for the construction of the accumulator.

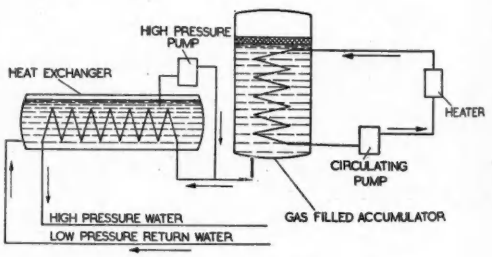

Fig. 2

Another type of gas-filled accumulator is shown in Fig. 2. It is usual practice to install for hydraulic presses a central accumulator plant connected with the individual presses by a high pressure pipe system operating at 200-250 atm. pressure, the accumulator capacity ranging from 500 to 1500 litres of water. For plants of this type ethylene may be used as working gas, the vapour pressure of which amounts to 50·8 atm. at 10 deg. C. In this case the required accumulator pressure of 200-250 atm. will be obtained by heating the accumulator to 80-100 deg. C.

A LIQUID-PROPELLANT ROCKET MOTOR.

By LOVELL LAWRENCE, JR. (From *Mechanical Engineering*, Vol. 68, No. 7, July, 1946, pp. 615-616, and 636, 5 illustrations.)

THE rocket motor, as distinguished from current aircraft power plants of the jet-propulsion or propeller type, does not operate on atmospheric oxygen, but depends upon a supply carried with the motor. The oxygen is furnished by an oxygen-yielding compound, known as the oxidizer ; the fuel may be any hydrocarbon, both of these being known as the propellants.

The jet-reaction engine under consideration utilizes liquids for propellants which are fed into the combustion chamber, and for this reason its operation cannot be compared to the powder or solid-propellant rocket motor whose entire charge of fuel is lodged in the combustion chamber. The liquid-propellant motor has been widely developed in both the United States and Germany because it can be repeatedly operated for long periods of time by merely replenishing the propellant supply.

CONSTRUCTION OF MOTOR

Basically, this rocket motor, Fig. 1, consists of an injector, the counterpart of the reciprocating-engine

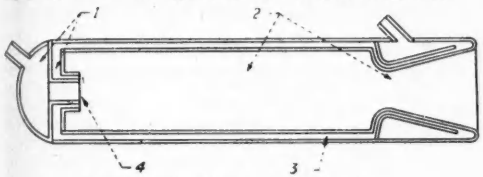

Fig. 1. Basic scheme of rocket motor.

carburetter, a combustion chamber, nozzle, and cooling jacket. The injector indicated at 1 is made up of two chambers, one to feed the oxidizer into the injector jet, the other to feed the fuel. An igniter or starter, not shown here, is usually incorporated in the injector when the propellants used are not spontaneously combustible. The combustion chamber and exit nozzle are usually considered as a unit, as shown at 2, since the entire combination must be cooled with one of the propellants flowing in the space formed by the outer jacket and baffle, 3. It should be noted that since this cooling arrangement returns the dissipated heat energy to the combustion chamber with the propellant, there are practically no cooling heat-rejection losses, thus affording a regenerative power plant.

After the propellant has travelled up the cooling jacket, it then enters the space at the entrance to the injector jet, and finally enters the combustion chamber, impinging on the other propellant which has entered at 4. It is possible to use as a coolant either the oxidizer or the fuel, depending upon their characteristics.

In the combustion chamber the fuel and oxidizer are burned, thereby converting thermal energy into high-pressure gases which are ejected from the exit nozzle. The exit nozzle changes the internal energy of the rapidly expanding combustion gases into kinetic thrust energy by accelerating the gases to supersonic velocities as they move from the high-pressure region in the combustion chamber to the relatively low-pressure region outside of the nozzle. Critical factors in nozzle design are the throat area and exit flare, the throat area being the major factor since it controls the rate of ejection of the gases. The exit flare angle is designed to follow the natural expansion of the gases as they pass from the nozzle throat to the atmosphere.

THEORY OF REACTION PROPULSION

The propulsive action of the rocket motor is described by the fundamental impulse-momentum equation, $F = \frac{mV}{t}$, solved for the resultant force in terms of the change of momentum of the propellants relative to the motor. The thrust force F, is measured in pounds ; the propellant mass m, in slugs ; the velocity of exhaust gases V, in feet per second ; and the time interval t, in seconds.

In considering the economy of operation of the liquid-propellant rocket motor, it is advisable to keep in mind the distinguishing characteristics of the motor as previously outlined. Whereas fuel consumption is the yardstick of economy of the air-stream and internal-combustion engines, propellant consumption is the basic measure used for the rocket motor.

The efficiency problems of the rocket motor are the same as the conventional propeller-type power plant where the product of the thermal efficiency and propulsive efficiency is equal to the over-all efficiency. The thermal efficiency of both power plants is a function of the available energy for propulsion after the engine has converted the thermal energy of the fuel into output shaft horsepower or exhaust-jet energy as the case may be. The propulsive efficiency is a measure of the ability of the propeller or exhaust jet to convert available shaft or combustion energy (respectively) into propulsive effort.

In practice, the thermal efficiency may be determined by substituting measured thrust and propellant flow rate in the fundamental impulse-momentum equation $F = \frac{mV}{t}$ and calculating the effective jet velocity. Then the square of the ratio of effective jet velocity to theoretical jet velocity is the thermal efficiency. The theoretical jet velocity is based upon the available B.T.U. content per pound of propellant mixture. The mixture used in this instance is ethyl alcohol (75 per cent) and liquid oxygen. It is important to note that at the present time we are able to turn but 28 per cent of the available B.T.U. content into effective energy because of the molecular dissociation and cooling problem at high combustion-chamber temperatures. The thermal efficiency does not change with the velocity of the rocket but does improve with increasing altitude because of reduced atmospheric back pressure.

The propulsive efficiency is a function of the thrust, the velocity of the rocket relative to the ground, the net thermal energy, and the kinetic energy of the propellants by virtue of their motion relative to the ground. Fig. 2 shows a graph of the propulsive efficiency versus the ground speed of a rocket for a jet velocity of 6600 ft./sec. (4500 m.p.h.). It will be noted that as the ground speed approaches the velocity of the jet, a maximum efficiency of 100 per cent is obtained.

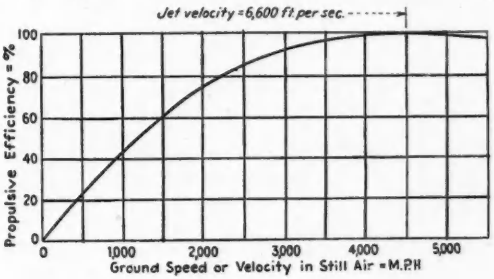

Fig. 2. Propulsive efficiency versus speed.

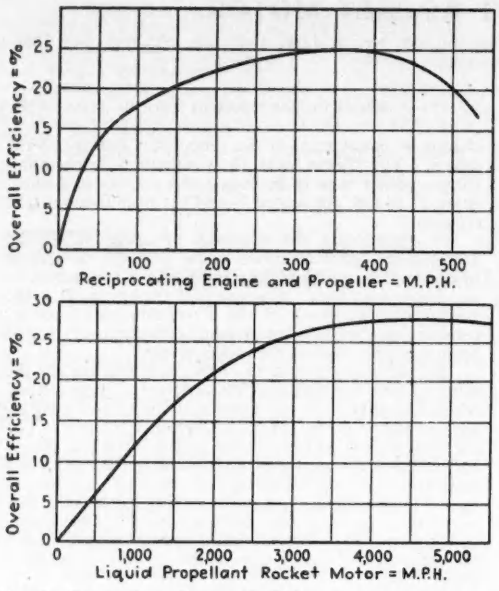

Fig. 3. Comparison of over-all efficiency versus speed for reciprocating engine with liquid-propellant rocket motor.

Fig. 3 shows a comparison of the over-all efficiency versus speed for a conventional engine with propeller, as compared with a liquid-propellant rocket motor. The efficiency scale is the same for both. The velocity scale of the rocket motor is 10 times that of the reciprocating engine. Surprisingly enough, the rocket motor has a better over-all efficiency than the best efficiency of the conventional type when it is operating in the proper speed range near its exhaust-jet velocity. This is possible because 100 per cent propulsive efficiency is attainable for the rocket where the aeroplane propeller utilizes about 80 per cent of the available shaft power. Since the rocket motor has a 25 per cent efficiency at 270 m.p.h., it can get the same mileage per pound of propellant mixture as the conventional-engine mileage per pound of fuel at 350 m.p.h. Therefore the rocket can travel the same distance in about 10 per cent of the time.

ROCKET MOTOR IN EARLY DEVELOPMENTAL STAGE

Let us take a look at a 1500-lb. thrust unit performing on a test stand. This 25-lb. unit compares with a 1000-h.p. aircraft engine. The peculiar "beaded" nature of the exhaust flame is caused by the reflection of the burning gases from the relatively motionless surrounding envelope of air. Since the jet stream is travelling at about 6 times the speed of sound in the adjacent air, the air is unable to compress rapidly enough and in effect forms a " solid " cylindrical tube through which the exhaust gases move. Eventually, friction slows the outer layers of the flame column down and air mixes with it causing a conical taper at the end of the flame.

In closing, we might mention an outstanding adaptation of the liquid-propellant reaction motor to conventional aircraft in order to emphasise the fact that rocket power has progressed from the experimental stage to a practical type of power plant that is now at hand.

The application we have in mind is the use of the liquid-propellant rocket for assisting take-offs of heavily loaded aircraft whose motors are not powerful enough to supply the necessary thrust. Very shortly the liquid-propellant rocket motor will emerge as a full-fledged aircraft power plant. The age of rocket power is here.

RUSSIA

CONCRETE AND CEMENT.

" The Work of Scientific Research Institutes."

By P. I. Gluzhge. (From *Gidrotekhnicheskoye Stroitelstvo*, No. 4, April, 1946, pp. 27-28).

1. TESTING MORTAR FOR USE ON UNDER-WATER BRICKWORK.

For underwater brickwork it is not sufficient, as is usually done, to judge the mortar only by its composition and its behaviour under the trowel. For best results it is essential to test for water separation and for deformation under load. A very simple method is suggested in each case.

(*a*) To test for water separation a glass jar 3 in. diameter by 7 in. long is filled with the mortar, stood upright, turned upside down, turned upright again, etc., every 10 minutes. Streaks of water along the walls reveal that separation has begun, and a good cement should not show this before 30 minutes.

(*b*) Deformation under load can be tested by standing a 1½ in. diameter by 1½ in. hollow cylinder on an air-dried brick, filling it with the mixture in question and, after removing the mould, loading the little column of cement with a second air-dried brick. Deformation is then measured every three minutes and when it ceases the specimen is inspected as to evenness of flattening, cracks round the circumference, pieces breaking away, etc. For a given consistency of the original mixture a big deformation without cracks or other defects is proof of good quality.

2. USE OF BROKEN CONCRETE AS AGGREGATE.

As a result of the war it is often tempting to save time and transport by using concrete demolition debris as aggregate in the re-built structure. Special tests in connection with the Dnieper Dam have shown that this is practicable and give the following conclusions :—

(1) Results are only good if the original concrete is as goods as the one it is desired to make (only if a very poor quality is acceptable it is possible to use material of up to 10 per cent. smaller ultimate).

(2) The re-utilization, instead of sand, of the finer particles demands an undue increase in the proportion of cement and is not recommended.

(3) The relation between strength and the water/cement ratio holds good, but the actual value of this ratio for a given strength is lower than with ordinary aggregate. Therefore, the mixtures must be determined, not from the usual formulæ, but on the basis of experiments with the actual aggregate to be used.

(4) The specific gravity is somewhat lower with the artificial aggregate and must be determined experimentally.

(5) The amount of cement to be used per cubic yard of concrete can be reduced in spite of the reduced water/cement ratio. To effect this the aggregate must be moistened, but not saturated, to improve its mobility. Preliminary experiments will indicate the optimum amount of moistening, giving mobility without appreciable reduction in strength.

(6) Given equal crushing strength, the bending strength is somewhat better when artificial aggregate is used.

(7) Concrete made with artificial aggregate loses its mobility very much more rapidly, but fills the mould very well in its " thickened " state if vibrators are used.

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Cations							
Calcium	Ca	3.2	-	9.4	-	10.7	-
Magnesium	Mg	0.8	-	0.36	-	1.09	-
Sodium	Na	0.46	0.23	1.0	0.31	1.66	0.44
Total		4.46	0.23	10.76	0.31	13.45	0.44
Anions							
Carbonate	CO_3	4.2	0.24	12.4	0.29	10.5	0.57
Chloride	Cl	1.8	0.06	2.5	0.12	2.84	0.30
Sulphate	SO_4	1.35	-	3.48	0.03	11.95	-
Nitrate	NO_3	-	-	-	-	1.15	-
Total		7.35	0.30	18.38	0.44	26.44	0.87
Total ions in solution		11.81	0.53	29.14	0.75	39.89	1.31
COST per 1000 gallons		5.22d		9.83d		16.5d	

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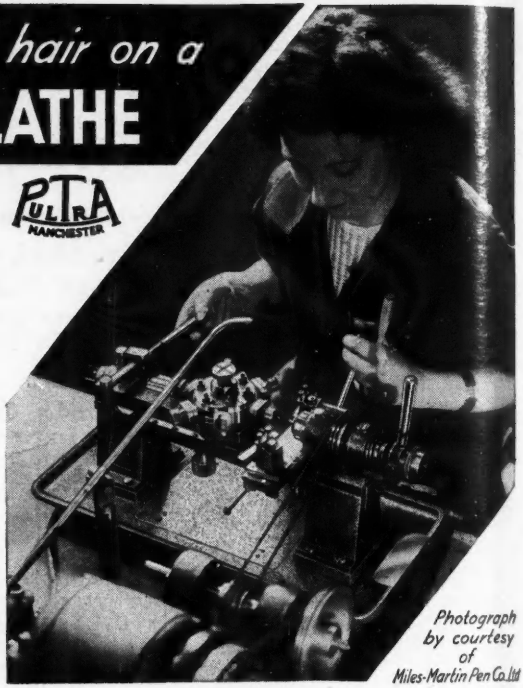

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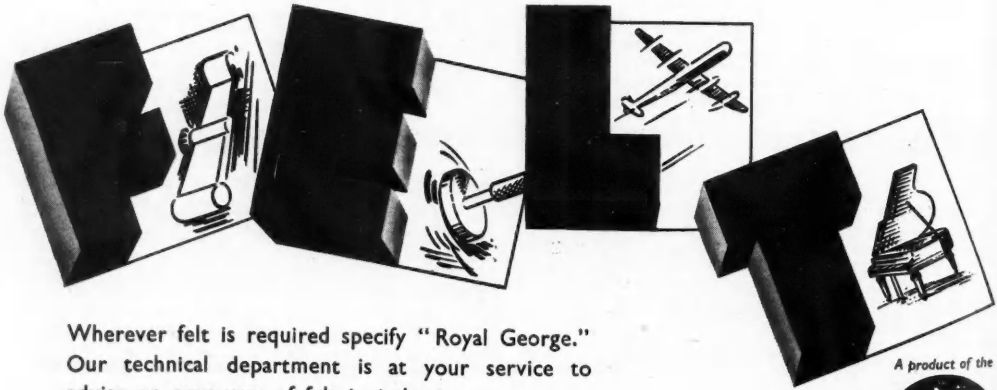

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CLASSIFIED ABSTRACTS

Numerous important articles which, for lack of space, cannot be published in full in our main editorial columns, are regularly abstracted in this section. Subscribers may obtain photostat copies of all original articles at cost.

DIESEL ENGINES

The Use of Heavy Oils in Stationary Diesel Engines

By M. ZWICKY AND H. RINGGENBERG. (From *Technische Rundschau Sulzer, Switzerland*, No. 2, 1946, pp. 1-5, 7 illustrations.)

THE development of air and land traffic based on highly refined light oils, and the introduction of new refining processes, make it imperative to use the cheaper residue oils for stationary Diesel engines. With long running times, the fuel costs form the greatest part of total running expenditure, and it pays to admit increased initial costs if cheaper fuel can thus be used.

Low speed engines with relatively large cylinder dimensions are best suited for heavy oils of low combustibility, as oil jets tend to impinge on the cylinder walls leaving solid unburnt residue. On the other hand, incombustible fractions (sediments) of the fuel always form solid residue, wearing and smutting up the cylinder faces. Erosion through refinery sediments, transport dust and steel particles, as well as corrosion through sulphur, distillation residues and perhaps sea water, contribute to cylinder wear which may well be double that of a light oil engine and which may rise appreciably higher with engines unsuited to heavy oils. Lubrication, although important, plays a minor part compared with the effect of fuel oil.

Engine design can minimize the disadvantages of the use of heavy fuel and counteract the influence of low inflammability, corrosive and erosive wear. Main design points are: high compression heat, high cylinder wall temperature, low speed, correct dimensioning of injectors and combustion spaces, special alloys for cylinder linings, filtering and pre-heating of fuel, magnetic filtering of lubricating oil. Pre-compression has proved very helpful, in one special case reducing wear by 50 per cent.

Sulzer engines are specially designed for running on heavy oils, with relatively large cylinder bores, low speeds, special cast alloys for linings and piston rings. "Combustion rings" in the compression chamber are air insulated from the cooled walls and remain so hot that impinging oil is burnt without leaving any deleterious residue. Special or magnetic filters or centrifuges are used according to fuel quality. The fuel is pre-heated by exhaust gases (up to 150 deg. C.) and the fuel pumps thus protected.

HEAT PUMPS

Service experience with heat pumps for central heating of the Zür ch Town Hall office buildings.

By A. OSTERTAG. (From *Schweizerische Bauzeitung*, Vol. 128, No. 12, September 21, 1946, pp. 147-150, 4 illustrations.)

WITH the extension of the Zürich town hall offices in 1943, a heat pump plant was added to the existing central heating boiler plant. Service performance during two winters (1944-46) can now be compared with the anticipated project performance and shows very satisfactory results. Meteorological statistics served to determine the degree of plant utilization, and plant efficiency was measured by wattmeters for electric energy input and a "Siemens" heat meter for heat energy output. This meter multiplies the quantity of hot water for central heating by its temperature drop on passage through the radiators. The winters were milder than anticipated, room temperature was kept at 63·5 deg. F. (65·5 deg. F. anticipated), the plant was operated with as few compressors running as possible, during as long a time of the day as permissible, with initial hot water temperatures closely adapted to heating requirements (125·5 deg. F. maximum). The plant was thus capable of covering total heating requirements even at temporarily very low outside temperatures of 16 deg. F. Anticipated heating efficiency of 13600 B.Th.U. per kWh was reached during the first winter and surpassed during the second winter when the river water remained warmer (38·5 deg. F.) than anticipated (34·5 deg. F.). The boiler plant was only used during pump service interruptions and supplied 2 per cent (anticipated 20 per cent !) of total heating requirements during the winter. Service and maintenance costs were considerably lower than for boiler heating and can be reduced still further by restricted working on Saturdays and by the use of automatic controls. The difficult coke position during the war years increased the economic advantages of the plant, but even under normal conditions, the plant could be amortised in 8 to 10 years.

The profitableness of a plant must be specially calculated for every case. Installation costs can be reduced by using smaller evaporators if heat transfer is improved, a higher average river water temperature assumed, or if under-ground water can be used. Service costs depend on electricity as compared with coal prices; however, the heat pump runs efficiently even for very short periods whilst a boiler is inefficient during heating-up and when running on small loads. Savings increase with larger well distributed heat requirements; the Zürich plant shows unfavourable conditions in this respect as working periods are short, river water temperatures low, and because the relatively small radiator surfaces in existing buildings require a high initial hot water temperature. Completely new plants will be more economic if they can be worked on part load during the whole year (e.g. in connection with laundries, industrial or refrigeration plant, etc.), if ample radiator surfaces are provided, or if radiation heating is installed with a maximum initial hot water temperature of only 104 deg. F.

METALLURGICAL ENGINEERING

Determination of the Depth of Surface Decarburization in Rolled and Drawn Steel.

By H. KJERRMAN. (From *Jernkontorets Annaler, Sweden*, Vol. 130, No. 7, July, 1946, pp. 239-252, 20 illustrations.)

As a preliminary to standardization the author has made a survey of different methods in use for determining the surface decarburization of steel. In practice, the most appropriate method is to measure the thickness of the decarburized surface layer microscopically on an etched cross section.

It may, however, admit of some question, to which depth the decarburization should be measured, and the author gives three alternative definitions:

a = surface decarburization layer, in which the structure consists of ferrite, i.e. where the carbon content is practically nil.

b = surface decarburization layer, in which the structure may vary from 100 per cent ferrite to 100 per

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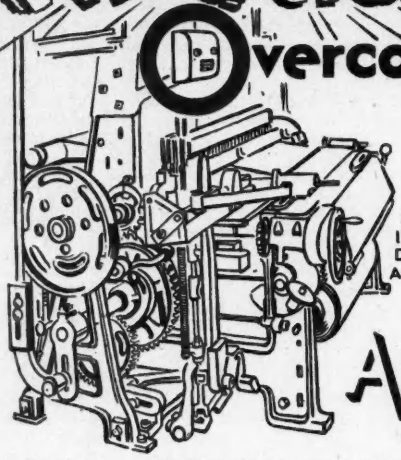

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cent lamellar pearlite, i.e. up to eutectoid composition.

c = surface decarburization layer, in which the structure indicates a lower carbon content than the average of the steel.

Examples of measurements according to these three definitions are shown in micrographs.

Influence of Structure and Composition of Alloys on their Mechanical Properties.

By Y. B. FRIDMAN. (From *Vestnik Inzhenerov i Tehnikov, Russia*, No. 2, 1946, pp. 44-50, 15 illustrations.)

THE author analyses the influence of such factors as grain size, distribution of solid solutions, intercrystalline brittleness and tempering temperatures on the mechanical properties of metals under varying conditions, including extreme low temperatures. He treats the latter in a novel manner, using a " diagram of mechanical condition " which reflects :—

(*a*) type of failure—tension and shear.

(*b*) stress conditions.

(*c*) a complete picture of the flow of metal as distinct from measuring the effect of a particular plastic deformation (as in Brinell and other tests).

Unfortunately this article by itself, while giving a great many very interesting curves, is insufficiently self-explanatory as to the author's obviously very interesting methods of avoiding the pitfalls of judging metals behaviour by such arbitrary and often misleading characteristics as " ultimate tensile ", " Brinell ", and " percentage elongation of standard specimen ". From the way the author assumes that his methods are understood it would appear that they have gained considerable recognition in Russia, but as their development covers the war years it is doubtful whether they are known in this country. Thus the article is very hard to follow though some of the conclusions are interesting.

For instance, it is shown that toughness and ductility of metals are drastically reduced by a coarse-grained structure only if the stresses are such as to induce a tension type of failure, whereas the tendency to fail in shear or in any other way which is dependent on shear is barely affected.

It is also shown that the idea that increased concentration of solid solutions reduces ductility is only correct for alloys in a ductile condition. Quite the opposite is true for alloys in brittle condition. Thus, while the addition of 5 per cent nickel to iron slightly reduces its ductility at + 20 deg. C. it increases it enormously at — 195 deg. C. (true reduction of area increases from about 2 per cent to over 50 per cent).

LITERATURE :—

Y. B. FRIDMAN, *Deformatsia i Razrushenie Metallov pri Statícheskih* ; Oborongiz, 1946. *Journal Tehnicheskoi Fiziki*, XI (1941), No. 10, IX (1939), No. 16 and X 1940, No. 16.

E. WELLAUER, *Metal Progress*, Vol. 37 (1940), No. 6.

E. BAIN, *Functions of Alloying Elements*, 1939.

A. BURKHARDT, *Technologie der Zinklegierung*, 1939.

METALS CORROSION

Some Aspects of the Corrosion of Aluminium.

By P. F. THOMPSON. (From *Journal of the Council for Scientific and Industrial Research, Australia*, Vol. 19, No. 2, May, 1946, pp. 157-165, 3 illustrations.)

THE corrosion of aluminium and its relation to film formation is discussed. Breakdown of the film is due to several causes. Its repair is effected by the products of two types of corrosion—" differential aeration " and " hydrogen evolution ". It has been found that aluminium is also corroded by copper ion discharge and also that this initiates continuous hydrogen evolution.

By electrochemical measurement, film growth and breakdown have been traced. The calculated limit of hydrogen evolution was found to coincide with definite inflections of the potential-temperature curves. Charts showing these relations are presented.

In confirmation, visual evidence of the actual evolution of hydrogen on abrased aluminium and magnesium in neutral liquids has been obtained. The importance of the effect of minute " chafing " or " fretting " movements on the stability of machines and structures is indicated.

More recently a continuous abrasion apparatus has been set up whereby more negative potential values than those shown on the chart have been observed, and the indication has been given that two types of film may exist, namely, the stabilized and the unstabilized.

PLASTICS

Relation between the Impact and Flexural Tests for Moulded Plastics.

By L. E. WELCH AND H. M. QUACKENBOS, JR. (From *Transactions of the A.S.M.E.*, Vol. 68, No. 5, July, 1946, pp. 547-556, 9 illustrations.)

A theoretical treatment of the phenomena of impact is presented which indicates that rupture under dynamic stress-strain conditions results from the same basic considerations as those governing static or flexural failure. This thesis is proved experimentally by reducing the problem to one of stress rather than the usual method of energy evaluation. When this is done with the aid of newly developed high-speed electronic techniques, it is conclusively demonstrated that impact loading is merely a special case of static loading, whereby the stress-strain conditions are identical except for a slight elevation in strength under dynamic conditions. It is shown that the existing methods of impact-strength evaluation such as the standard Izod and Charpy tests lead to erroneous results since the machines in use are apparently of too light a construction, and the technique is oversimplified. It is suggested that the static flexural-test technique, when properly carried out to evaluate the energy required for fracture and the notch sensitivity, is sufficient to predict the service characteristics of essentially brittle materials under conditions of both static and impact loading.

RADIO ENGINEERING

New Russian Radio Receiving Sets.

By V. V. ZENCHIKOV AND OTHERS. (From *Vestnik Electropromyshlennosti, Russia*, No. 5-6, 1946, pp. 12-21, 19 illustrations.)

UNDER a single heading the above journal publishes a series of very informative articles giving full details of post-war models " Leningrad " (" First Class Superhet "), " Rodina " (medium battery set), Motor-car receiver " A—695 ", " V.E.F. Super M—557 " (a medium quality model), and " Record " (a " popular " model for the masses).

In each case except the last a full diagram is given and to some extent commented upon and a number of interesting performance curves and tables are also included.

Of particular interest is the article concerning " A—695 " in which some of the technical difficulties of vehicle sets and the method of overcoming them are discussed at length It would be interesting to see whether this set is in fact, as it appears to be, more thoroughly adapted for road work than the usual run of such receivers.

STEAM POWER PLANTS

Steam Separation and Purification in Boiler Drums.

By P. B. PLACE. (From *Combustion, U.S.A.*, Vol. 18, No. 1, July, 1946, pp. 35-38, 6 illustrations.)

THE process of separation and purification of steam in a boiler is accomplished within the boiler drums by the

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Steam Power Plants—*continued.*

aid of baffles and other devices collectively known as drum internals. Separation and purification is a sequence of two or more steps, the principal steps being: (1) Primary separation, (2) Steam washing, (3) Steam drying. It is desirable that the steam leaving a primary separator should contain less than 5 per cent and not more than 10 per cent moisture. To reduce the 5 lb. to 15 lb. of circulating water delivered to the separator with the steam to less than 0·1 lb. in the steam leaving the separator requires separator efficiencies from 98 per cent to 99 per cent.

The steam leaving a primary separator is delivered to a steam washer or directly to the steam dryer. There are three phases to the steam washing process. First, there is the substitution and dilution of dirty boiler water by clean feed-water. Second, there is a simple mechanical elimination of boiler water by contact with the washer in the same manner as separation is effected by a baffle plate. Third, there is condensation in the washer which is inherently a de-aerating feedwater heater. The performance of a steam washer depends upon the following factors: (1) Solids concentration in the feedwater, (2) Solids concentration in the boiler water, (3) Quality of the separated steam entering the washer, (4) Quality of the washed steam leaving the washer, (5) Amount of purely mechanical separation that occurs in the washer.

The steam from a separator or washer is finally passed through a dryer to remove the last traces of moisture and dirt. The separating efficiency of a dryer should be such that the outlet steam will be of practically constant purity over the operating range of the boiler, up to the point of overload. At low pressures the average velocity of the steam passing through the dryer may be as high as 8 to 10 ft. per second, but at high pressure this velocity may have to be reduced to less than 2 ft. per second. An important factor in dryer performance is the amount of residual moisture in the steam delivered to the dryer. A drop in dryer efficiency because of leakage from 99·5 to 98 per cent makes a difference between satisfactory and unsatisfactory outlet steam purity.

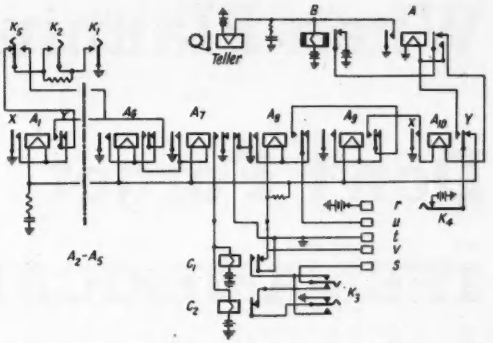
TELEPHONE AND TELEGRAPH ENGINEERING

Device for measuring relay times.

By S. SVANES. (From *Elektroteknisk Tidskrift, Norway*, Vol. 59, No. 20-21, July, 1946, pp. 267-270, 5 illustrations.)

THE Bergen Telephone Company have developed a device for measuring relay times which embodies a novel working principle. By mounting this device in a carrying case, it can also be used for the time study of series of relay operations in actual working circuits on site. The circuit of this apparatus for measuring relay times is shown in the accompanying illustrations. Here ten relays are seen to be connected in such a way that their contacts are closed in sequence when a starting switch is closed. The closing of relay No. 10 in turn operates a counter, while at the same time the nine other relays are opened. Shortly after that, relay No. 10 is also opened so that the relays are closed again in sequence, and so on.

A stop watch is started simultaneously with the starting of the relays and the time of one operating cycle is accurately determined by counting a large number of consecutive cycles, for instance 500 cycles. If one of the ten relays, as for instance relay No. 7, is put out of action by arresting it, then the time required for 500 cycles will be shortened. If for instance the time for 500 cycles was found to be 91·52 seconds with all ten relays in action, and if the cycle time is 87·30 seconds with only nine relays being operative, then the difference in cycle time is 91·52—87·30 = 4·22 seconds, and the



time required for the closing of one relay, that is, relay No. 7, therefore is

$$\frac{4{\cdot}22}{500} = 0{\cdot}00844 \text{ seconds} = 8{\cdot}44 \text{ milli-seconds.}$$

The circuit shown in the accompanying diagram is so arranged as to meet all practical requirements. Here K_1 is the starting key, while the keys K_4 and K_5 permit to vary the time interval between counts. The relay C_1 closes its front contact simultaneously with the breaking of the contact relay C_2. The terminals r, u, t, v, and s are connected with the circuits or contacts of the relay to be investigated.

TRAFFIC CONTROL

Street Traffic Signal Plant with Impulse Coils.

By N. FORCHHAMMER. (From *Ericsson Review, Sweden*, No. 1a, 1946, pp. 81-84, 7 illustrations.)

IN recent years a system of street traffic signal plant has been introduced in which the signals are wholly or partially operated by the vehicles themselves. An essential feature in this plant is the special impulse device, the impulse coil. In place of a street contact, where a mechanical action by the vehicle is converted to electrical contacts, a purely electrical inductive action on a buried coil is employed. As a motor vehicle passes, its steel mass acts on the earth field through the coil, inducing a current in it which actuates the connected impulse relay. By way of an accessory relay this operates the remainder of the signal mechanism. The sensitivity can be so regulated that either vehicles and cycles or vehicles only will actuate the relay.

One advantage of the impulse coil is that no movable parts lie in the roadway; the coil is embedded in concrete, after which it is covered by the surface material of the road. Thus there is no possibility of rain penetrating it, or snow or ice affecting it. As the whole of the energy serving the relay is produced inductively, there is normally no tension in the coil, so that no insulation troubles arise with the parts embedded in the road. There is no wear on the parts in the roadway; the coils lie 4-6 cm below the surface of the road, which can be relaid without disturbing them. The fitting is simple and subsequent adjustments are not required should the roadway settle or be displaced; for mechanical contacts, on the other hand, a firm foundation must be provided and the part above must be accurately adjusted to the contact.

The chief feature of the system is the sensitive relay which receives the induced impulse from the coil. It is constructed with a moving coil system and a special contact device. All the other auxiliary and time relays in the apparatus operate with 12 volts, supplied by a built-in rectifier. The contacts for the lamp circuits are served by a motor-driven control apparatus. The time-setting of the control apparatus is done by resistances.

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VIBRATIONS

Calculation of the Multiple-Span Critical Speeds of Flexible Shafts by means of Punched-Card Machines.

By A. W. RANKIN. (From *Journal of Applied Mechanics* (*A.S.M.E.*), Vol. 13, No. 2, June, 1946, pp. A117-A126, 15 illustrations.)

IN turbine design, the critical speeds must be adequately spaced from the normal running speed to assure a smoothly operating unit. It is essential, therefore, that these critical speeds be predicted early in the design. Several accurate and general methods of calculating multiple-span critical speeds of irregular shafts are available in the technical literature, but in the author's experience these methods, while completely competent in a technical sense, are not satisfactory for routine design because of the discouraging number of calculations which must be made. To overcome this difficulty, the author has devised a method employing standard punched-card business machines, now in relatively wide use in machine accounting. The data under study are fed into these machines by means of cards which are punched in accordance with the known data, and the machines are actuated by contacts through the punched holes. The results of the operations performed by the machines are punched automatically on cards in a similar fashion.

WATER RESOURCES

Scientific Problems of Water Engineering of the U.S.S.R. in the Fourth Five-Year Plan.

By A. N. AKHUTIN. (From *Izvestia Akademii Nauk, Russia*, No. 2, 1946, pp. 177-184.)

THE 1946-50 Plan involves a great number of hydraulic developments on a very large scale, covering hydro-electrics, inland navigation, irrigation, drainage and water supply. The principal ones are as follows :—

The Molotov Development on the river Kama : 500,000 KW and improvements to river navigation. This to be followed later by an associated development at Solikamsk (400,000 KW) and intercommunication between Kama, Pechora and Vychegda river basins.

Angara River Development. This river drains the vast Baikal lake so that when fully developed it will yield 9,000,000 KW.

Development of River Oka. This will be effected in two stages with plants at Kaluga and Kolomna. The former, which will be built first, will have an installed capacity of 150,000 KW. Improvements to navigation will again be incorporated.

River Kura. A 300,000 KW installation at Mingechaur will control floods, supply power for the Baku oil region and provide irrigation facilities for 2½ million acres.

Lake Sevan Project in Armenia is rather unique in that it will utilise the water of this mountain lake (6300 ft. above sea level, 560 sq. miles area) at a rate beyond the rate of replacement. Hydro-electric plants to a total of 575,000 KW capacity will draw down the lake level by 160 feet in the course of the next 60 years. By that time the surface area of the lake will be so much reduced that the reduced rate of evaporation will result in improved natural outflow.

Irrigation and Hydro-Electric Projects in the Syr-Darya River Basin will involve a total of over 20 power stations of which 8 are already in operation and 5 under construction.

Reconstruction of the Volga-Baltic canal system to a minimum depth of 8 ft. 3 ins.

Drainage of White Russian Marshes along the Pripet river, to bring under cultivation 250,000 acres of new land.

Water Supply Developments on a very large scale to the Donetz, Kazakhstan, Ural and Kuzbass industrial regions.

WELDING

Influence of grain size of flux on high speed welding.

By D. M. RABKIN AND B. I. MEDOVAR. (From *Avtogennoe Delo, Russia*, No. 1, 1946, pp. 11 & 12.)

THE authors dispute the theory put forward by I. M. Keiz (*The Welding Journal*, No. 3, 1943) and the idea commonly held in America that large flux grains result in better penetration and smaller fillet. They maintain that, for welding speeds of 1 ft./min. and over, the reverse is true.

They were particularly interested in speeds of 3 to 5½ ft./min. and experimented with flux in the form of : (*a*) dust ; (*b*) normal powder (sieve with 64 holes per sq. inch) and ; (*c*) large-grain (sieve with 25 holes per sq. in.). The results were as follows :—

1. Voltage across arc increased as between (*a*) and (*c*) by 18-27 per cent.

2. Amount of flux used increases from (*a*) to (*b*) by some 35 per cent. and from (*b*) to (*c*) by more than 120 per cent.

3. With increasing grain size of flux the outer component of arc length grows very rapidly and so does the length of arc that can be maintained without breaking.

4. The width of weld increases with increased size of flux grains while its height and penetration decrease. As between (*a*) and (*c*) the changes were approximately + 40 per cent. and — 33 per cent. respectively.

It would seem that with large-grained flux the energy of the arc is directed far more to the melting of extra flux and far less to the melting of the metal, and while the resulting weld has superficial advantages, they are outweighed by the drawbacks. The authors therefore recommend normal grain size of flux for high-speed welding, and a note from the editor draws attention to a previous article by a different authority pointing to similar conclusions. (K. V. Lyubavsky in *Avtogennoe Delo*, No. 1, 1945, p. 12.)

The Operation of Electric Welding Installations and their Improvement.

(From *Promishlennaya Energetica, Russia*, Vol. 3, No. 4-5, 1946, pp. 4-5, 3 illustrations.)

AT the Uralmachine-Works the introduction of automatic electric arc welding has resulted in lowering the electric energy consumption from 4 KW hours per kg. of deposited metal to 2 KW hours. The execution of welds with high current densities with the use of coated electrodes has made possible the introduction of constant voltage welding at a voltage of 42 volts instead of 63 volts, as used hitherto. This measure has increased the efficiency of the welding plants from 32 per cent to 48 per cent, and has resulted in an annual saving of 1,500,000 KW hours. The replacement of multi-station welding transformers by single-station motor-generator sets has resulted in a further saving of 1,785,000 KW hours because of the reduction in idling time and of the plant efficiency achieved.

The introduction of automatic welding machines equipped with self-regulation devices manufactured on the premises, allows of the deposition of 600 tons of weld metal per annum with ten such machines, the corresponding power consumption being 1,200,000 KW hours.

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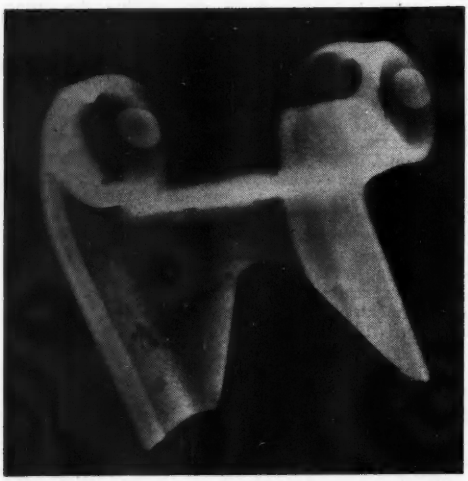

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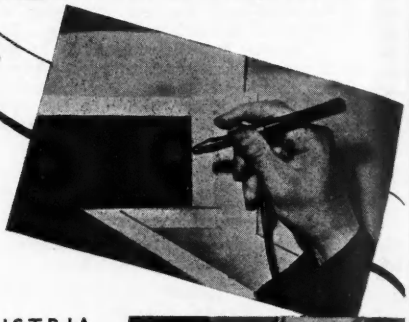

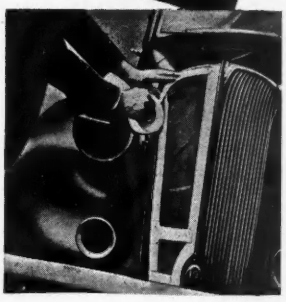

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PERSONAL.

Mr. J. H. Enion, a Director of the Igranic Electric Co., Ltd., and Managing Director of Metal Industries (Electrical Group), Ltd., has now joined the board of Metal Industries, Ltd.

Mr. W. C. Fahie, has been appointed Head of the Instrument Section in the Physics Department of the British Iron and Steel Research Association.

Mr. T. Hands, O.B.E., Managing Director of the Cosmos Manufacturing Co., Ltd., has been appointed a Director of the Edison Swan Electric Co., Ltd. **Mr. V. Plascott,** General Works Manager of Edison Swan Cables, Ltd., and **Mr. F. E. C. Miller,** Director of the Edison Swan Electric Co., have been appointed to the Board of Edison Swan Cables, Ltd.

Mr. Geoffrey Heyworth, Chairman of Lever Bros., and Unilever, Ltd., has been appointed Chairman of the Advisory Council for Scientific and Industrial Research, in succession to **Lord Riverdale,** who is retiring after holding the appointment for nine years.

Mr. J. J. Hickey has been appointed Manager Mechanical Goods Design of Goodyear Tyre & Rubber Co. (Great Britain) Ltd., Wolverhampton.

Mr. E. S. Little has been appointed to the Board of Associated Electrical Industries Finance Co., Ltd. This company is concerned with the financial operations of all the companies of Associated Electrical Industries, Ltd., including the British Thomson-Houston Co. In consequence of his new appointment, Mr. Little has retired from the Board of the British Thomson-Houston Co., Ltd., but he continues to be comptroller and Secretary of that company.

Mr. D. C. Lorkin and **Mr. S. G. Mundy** have been appointed Directors of Lancashire Dynamo and Crypto, Ltd.

Mr. C. F. Russell, General Manager of Specialloid Ltd., has gone to Montreal to finalise organization and administration in the Canadian subsidiary.

Mr. J. Clifford Wood, Commercial Manager, John Fowler & Co. (Leeds) Ltd., has been appointed a Director of the company.

NEW LITERATURE

Diamond Tool Progress.—"The Industrial Diamond Review" has issued a list of recent scientific and technical articles dealing with progress in respect of diamond tools. Reprints have been made from articles and data sheets published and are available to readers free of charge while the supply lasts. Apply to "Industrial Diamond Review," 226 Latymer Court, Hammersmith, London, W.6.

War Record.—Messrs. George Kent Ltd. of Luton have issued an illustrated summary showing the firm's principal war products and activities from September 1939 to August 1945. Messrs. Kent have also issued a miniature catalogue—the first post-war edition—showing their water meters and instruments used for water supply and sewage disposal, and flow meters and other instruments used in power generation, oil refining and in many other manufacturing and processing industries.

Report of Conference on Works Magazines.—Industrial Welfare Society, 14, Hobart Place, Westminster, London, S.W.1. 32 pp. Price 2s. post free.

It is estimated that before the war over 300 firms in Great Britain produced 'house' magazines. Many of these magazines ceased publication during the war, though a substantial proportion were continued, often as news sheets for members serving in the Forces. With the partial relaxation of paper restrictions many firms are to-day restarting their magazines and many new ones are being planned. The emphasis to-day is to give full information about the policies, technical developments, commercial activities of the firm and a fair share of social news as well. The works magazine has become an organ of industrial relations.

The Report of the Conference on Works Magazines, organized by the Industrial Welfare Society earlier in the year, discusses the scope and purpose of the employee magazine and details some of the practical problems of production. Appendices detail current costs, titles of existing magazines and give notes on present day trends.

New Electrical Resistance Alloy and Handbook.—Two booklets have been published by the Metals Division of Imperial Chemical Industries. One summarises the properties of "Kumanol," a new copper base alloy, remarkable for its very low temperature coefficient, and provides a concise practical summary for electrical engineers. It includes useful tables, setting out the dimensions, resistances and weights of tape and strip in the new alloy, and the current-carrying capacities of "Kumanol" wires of various gauges. The other is a handbook and deals with "Kynal" and "Kynalcore" and gives details of the physical properties of these wrought aluminium alloys.

The data is arranged in eight separate schedules so as to enable constructional engineers and designers to select the alloys most suitable for their specific requirements. One of these schedules sets out the general physical properties of aluminium and its alloys compared with those of other metals.

"Morgan's at War," published by the Morgan Crucible Co. Ltd., London, S.W.11.—An interesting brochure has been published by the Morgan Crucible Co. Ltd., showing the firm's achievement under fire during the years 1939-45. It is dedicated to employees who suffered at home and abroad by enemy action; to the firm's long service men and women—their prop in times of stress and difficulty, and customers at home and abroad who sympathised with their circumstances and were indulgent in short-comings. Interesting details of the firm's production, many good photographs, and charts prepared by the Isotype Institute, are presented in a visual form in the brochure.

The "Nickel Bulletin", Vol. 19, No. 6.—"The Nickel Bulletin" for June contains numerous abstracts, many of American origin, including a summary of reports on the adhesion of electro-deposits, nickel plating on aluminium and zinc-base die castings, foundry practice, standard test bars for non-ferrous castings, bearing metals, magnetic materials and methods of improving the forming qualities of low alloy steels.

Copies of the "Nickel Bulletin" may be obtained, free of charge, from The Mond Nickel Company, Limited, Grosvenor House, Park Lane, London, W.1.

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The effect of Type of Oven upon the Properties of the Coke.

"Experiments on Coking Practice."—A report just issued by the FUEL RESEARCH ORGANISATION OF THE DEPARTMENT OF SCIENTIFIC AND INDUSTRIAL RESEARCH, and published by H.M. Stationery Office, provides information of value to the coke-oven and metallurgical industries, particularly to those undertakings faced with the need to instal new batteries of ovens.

In the past, experience of the behaviour of coal coked in any particular battery of coke-ovens has generally been limited to coals or blends of coals drawn from an area adjacent to the battery. Shortly before the war, however, the Fuel Research Organisation of the D.S.I.R., in co-operation with the coke-oven and steel industries, began a comprehensive investigation of the carbonization of different coals in commercial coke-ovens of several types operating under various conditions. The objects of the investigation were to obtain precise information on the influence of the coal selected, the design of oven, and the methods of operation on the quantity and character of the coke produced, and the suitability of the coke for metallurgical purpose.

It was not considered desirable to draw definite conclusions from the limited experimental data available, but the following trends were noted. Increased rates of coking appear to reduce the size of coke and to reduce its strength. The coke from the narrowest oven was exceptionally large. The series of experiments indicated no orderly relation between width of oven or rate of heating, on the one hand, and coke abrasion indices or reactivity on the other. Coke reactivity appears to be governed more by the nature of the coal carbonized than by condition of coking.

In addition to tables and charts of results, the report includes a large number of photographs of samples of the cokes, and of sections specially prepared to provide information on their internal structure.

"Facilities for Tests on New Housing Developments."—By J. L. CARR, M.Sc., M.I.E.E. Published by The British Electrical and Allied Industries Research Association, 15, Savoy St., London, W.C.2. Price 2s. Postage 3d.

This brief report is intended to encourage the provision of facilities for conducting load tests in substations supplying housing developments of interest from the point of view of electricity supply.

It describes various arrangements which can be made at slight extra cost when erecting the substation, and which will enable suitable sectionalization to be made and measuring instruments connected whenever it is desired to obtain information.

BUSINESS NOTES.

D.C. BIAS STREET LIGHTING SYSTEM HELPED THE R.A.F.

A NOVEL war-time application of the D.C. Bias Street Lighting Control System introduced in 1937 by Standard Telephones and Cables Limited, has now been announced.

During the war, R.A.F. Stations grew in size and number, and many new buildings were erected to provide working, living and recreational quarters for R.A.F. and W.A.A.F. personnel. One result of this expansion was that the station emergency lighting plants were unable to handle the new loads, a very serious matter if failure of the supply mains coincided with operational needs such as airfield lighting. Faced with the prospect of trying to place rush orders for new and bigger engine-generator sets on our already overloaded manufacturing resources the Service Authorities decided instead to instal "S.T.C." D.C. Bias Control.

This aid enabled all non-essential circuits to be isolated from the station network by pressing a button in the Watch Office so that when the emergency plant was in action, essential needs could be met without overload problems. On restoration of the main power supply another push button enabled the non-essential circuits to be re-connected to the network. 118 R.A.F. Stations were fitted with this system, involving the supply of 8,000 receiving units.

STEEL FOUNDERS' CONVENTION.

The British Steel Founders' Association held its first Technical Convention at the Royal Victoria Hotel, Sheffield, from 3rd to the 5th October, 1946.

Some of the principal visitors who attended were: The Lord Mayor of Sheffield (Alderman C. W. Gascoigne, J.P.); The Master Cutler (Mr. Eric S. Osborn); Sir Charles Goodeve, F.R.S. (Director B.I.S.R.A.); Mr. W. Leonard (Parliamentary Secretary to Minister of Supply); Mr. D. Howard Wood (President, Institute of British Foundrymen); Mr. J. G. Pearce (Director B.C.I.R.A.); Mr. Vincent Delport (Manager, "The American Foundry").

After a series of Works Visits the first session opened with a discussion on "Sand Practice current in the Lancashire and Yorkshire Region".

The second session was a discussion on the "Relationship" between temperature and fluidity in Converter and Electric Steels", and the third session opened with an address by Mr. Basil Gray, Chairman of the B.S.F.A. Main Technical Committee on the past and future activities of the Committee.

SCIENTIFIC CONFERENCE.

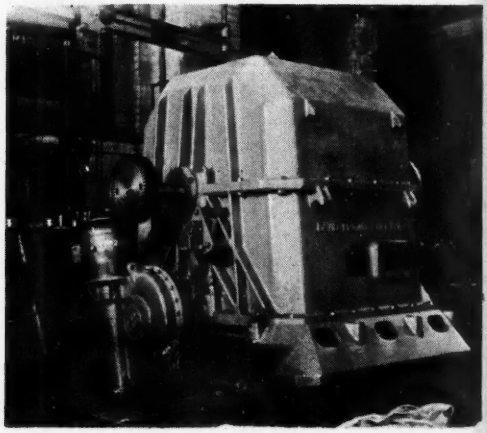
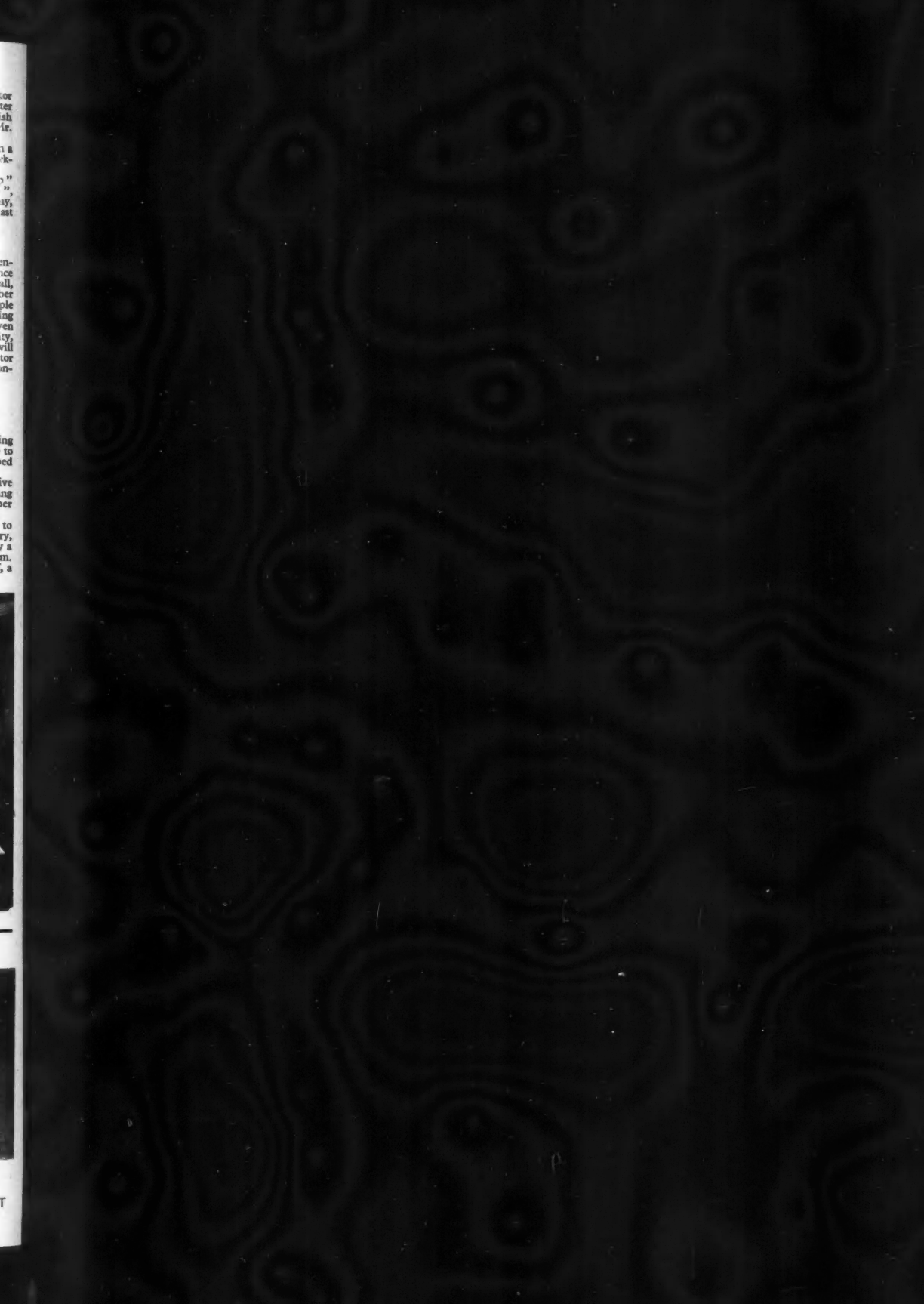
Science and the Public Welfare.—The Association of Scientific Workers announces that a conference, under the title of "Science and the Public Welfare," will be held at the City Memorial Hall, Sheffield, on the afternoons of Saturday and Sunday, November 16th and 17th. The object of the conference is to give the people of the Northern Counties some insight into the part science is playing in the interests of their general welfare. Addresses will be given by Professor P. M. S. Blackett, F.R.S., of Manchester University, and Professor J. D. Bernal, F.R.S., of London University, who will speak on the social implications of science. Dr. J. Rotblat, Director of Atomic Research at Liverpool University, is to address the conference on "The Future Use of Atomic Energy."

NEW EQUIPMENT.

David Brown & Sons Barring Gear.—An interesting application of a barring gear with a safety device has been made to a 32-inch centres co-axial double reduction helical gear unit, shipped recently by David Brown & Sons (Huddersfield) Ltd.

The unit, incorporating double helical gears, is designed to drive a heavy tube mill for cement grinding in South Africa. Transmitting 450 H.P., the speed reduction is from 735 to 23 r.p.m., and 100 per cent overload will be encountered on starting.

A separate double reduction worm gear unit is employed to rotate the mill when inspection, charging or re-lining is necessary, operating through the layshaft of the main gears. It is driven by a 9 H.P. motor, running at 2850 r.p.m., giving a mill speed of 0·25 r.p.m. and to make the operation of the auxiliary barring gear foolproof, a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